VŠB – Technical University of Ostrava

Faculty of Mechanical Engineering

Department of Power Engineering

DIPLOMA THESIS

Design of an Industrial Radial Air Fan

Návrh průmyslového radiálního vzduchového ventilátoru

Student: **Kannan Karupannan Gnanasundaram**

Supervisor: **Ing. Zdeněk Šmida, Ph.D.**

VŠB - Technical University of Ostrava
Faculty of Mechanical Engineering
Department of Power Engineering

Diploma Thesis Assignment

Student: **Kannan Karupannan Gnanasundaram**

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Description:

In the theoretical part of the thesis elaborate please the analysis on the topic of „Machines for energy transformation" with emphasis on dynamic bladed machines. Focus mainly on the working process of machines, energy transformation, basic parts of these machines and description of the working process using the enthalpy-entropy diagram.
In the practical part of the thesis please design the industrial radial (centrifugal) air fan with these given parameters: the flow performance 30 m3/s, pressure depression is 2 000 Pa, the single-stage fan will have the one-side suction, the rotation of fan will be 500 min-1, the suction air temperature is 40 °C, the suction air absolute pressure is 1 bar. Select the fan blades such that ideally 50% of the input work is transformed into kinetic energy and 50 % of the input work will be used to increase the pressure energy of the gas. The next of the necessary parameters please choose appropriately. In case of the modification of the given parameters from the reason of design optimization, please justify these modifications.
The design will consist calculation of velocities and angles for the design of blades, calculation of main dimensions of the rotor, calculation of electric input of fan and selection of suitable electric motor and geometric design of stator spiral chamber, schematic design of radial fan with the main dimensions and angles and technical drawings.

References:

MORAN, M. J., SHAPIO, H. N. Fundamentals of Engineering Thermodynamics – First Edition.. New York: John Wiley & Sons, 1990, 840 p. ISBN: 0-471-57117-2
YAHAY, S. M. Turbines Compressors and FANS – Second Edition. New Delphi: Tata McGraw-Hill Publishing Company Limited, 2002, p. ISBN: 0-07-042039-4.
LOGAN, E.Jr.: Handbook of Turbomachinery (Mechanical Engineering, No. 158) (2 ed.). New York: Marcel Dekker, 2003. ISBN 0-8247-0995-0.
DIXON, S. Larry; HALL, Cesare. Fluid mechanics and thermodynamics of turbomachinery – Fourth Edition. Oxford: Butterworth-Heinemann, 2013, 321 p. ISBN: 0 7506 7059 2.
SCHOBEIRI, M. T. Turbomachinery Flow Physics and Dynamic Performance. Berlin, Heidelberg: Springer, 2005, 522 p. ISBN: 3-540-22368-1.

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Supervisor: **Ing. Zdeněk Šmída**

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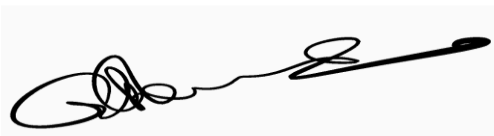

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ANNOTATION OF MASTER THESIS

Kannan Karupannan Gnanasundaram. *Design of an Industrial Radial Air Fan: Master Thesis*. Ostrava: VŠB – Technical University of Ostrava, Faculty of Mechanical Engineering, Department of Power Engineering, 2020, 72 p. Supervisor: Ing. Zdeněk Šmida, Ph.D.

My master thesis deals with the design of an industrial radial air fan. The first part of the thesis deals with the types of energy transformation machines and its working. Then the type of compressors, the working of different types is explained and the working of a centrifugal compressor with enthalpy entropy diagram is explained. According to the task the correct choice of selecting the electric motor for the power consumption is done. Then the calculation of the main dimension of the design is obtained and was designed in SOLIDWORKS 2019.

Keywords: Energy transformation machines, Centrifugal compressors, radial fan, enthalpy-entropy diagram, design.

ANOTACE DIPLOMOVÉ PRÁCE

Kannan Karupannan Gnanasundaram. Návrh průmyslového radiálního vzduchového ventilátoru. Diplomová práce. Ostrava: Vysoká škola báňská - Technická univerzita Ostrava, Fakulta strojní, Katedra energetiky, 2020, 72 s. Vedoucí: Ing. Zdeněk Šmída, Ph.D.

Diplomová práce se zabývá návrhem průmyslového radiálního vzduchového ventilátoru. První část práce se zabývá typy strojů pro transformaci energie a jejich fungováním. Poté jsou popsány dynamické kompresory a práce potřebná pro jejich pracovní proces a je také vysvětleno fungování kompresoru v entalpicko entropickém diagramu. Dle zadání byla provedena volba elektromotoru pro vypočtený příkon. Také byly spočteny hlavní rozměry a celý ventilátor byl navržen v programu SOLIDWORKS 2019.

Klíčová slova: Stroje pro transformaci energie, dynamické kompresory, radiální ventilátory, entalpicko-entropický diagram, návrh.

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Variables	Abbreviation	Units
P	Pressure	[Pa]
V	Volume	$[m^3]$
R	Universal gas constant	[J.kg^{-1}.K^{-1}]
T	Temperature	[K]
a	Work done	$[j.kg^{-1}]$
c	Absolute velocity	$[m.s^{-1}]$
w	Relative velocity	$[m.s^{-1}]$
u	Peripheral velocity	$[m.s^{-1}]$
D	Diameter	[m]
r	Radius	[m]
b	Width	[m]
z	Number of blades	[-]
η	Efficiency	[%]
α	Angle between peripheral and absolute velocities	[°]
β	Angle between peripheral and relative velocities	[°]
σ	Pressure ratio	[-]
ρ	Density	$[kg.m^{-3}]$
$\dot{V}_d$	Flow performance	$[m^3.s^{-1}]$
τ	Blade shape factor	[-]
$\dot{m}_d$	Transported amount	$[kg * s^{-1}]$
K	Reaction factor	[-]
n	Speed	min^{-1}
λ	Loss factor	[-]
ψ	Pressure factor	[-]
ε	Coefficient constant number of blades	[-]
φ	Volume factor	[-]
i	Enthalpy	$[kJ/kg]$
s	Entropy	[kJ/kg.K]

1 INTRODUCTION

Centrifugal or radial fans are mechanical devices to move air or other gases at some angle. In industrial surroundings formation of acidic fumes, hot gases, abrasive dust particles, etc., produced are needed to be moved to avoid accidents. Centrifugal fans play a vital role in this kind of industry to enable movement of the large flow of gases. These are specially designed to work in different situations.

My thesis is focused on the Designing of an Industrial Radial Air Fan, which includes calculation of main dimensions of the rotor and stator. The thesis can be divided into two parts, the first one is the theoretical part and the second one is the calculation part. The theoretical part includes the explanation of machines for energy transformation, their categories with some examples. It also includes the compressors and its divisions in general and focused on the dynamic compressors with explaining the types of fans and the centrifugal compressors with its working process, stages of a centrifugal compressor, and ideal and real working process using enthalpy entropy diagram.

The second part of the thesis is the practical part, where the Design of an Industrial Radial Air Fan is done with the calculation of main dimensions. The calculation can be divided into three parts, the first is the calculation of the rotor which includes calculating the inner and outer diameter, number of blades in the rotor, design of the rotor blade. The second is the design of the stator, the calculation of stator is divided into 8 different cross-sections to attain increased flow cross-section. The final part is the calculation of power consumption of the electric motor and the suggestion of suitable electric motor in practical use by using the obtained power consumption.

Once the calculations are done the thesis is taken into designing the radial fan. I designed the manufacturing schemes for the individual parts and the assembly drawing of the radial fan is done with the help of design software SOLIDWORKS 2019 and AUTOCAD 2020 for schemes. Some parts are needed to be selected which are not calculated in the calculation and need to finish the assembly drawing which includes the proposal of materials for the design of the industrial radial fan.

2 MACHINES FOR ENERGY TRANSFORMATION

According to the first law of thermodynamics “energy can neither be created nor destroyed but can be transformed from one form of energy to another form of energy”. Energy transformation is the process of conversion of one form of energy to another form of energy [23]. It takes place in three types namely,

- Primary machines
- Secondary machines
- Tertiary machines

2.1 Primary machines

Primary energy transformation machines are which convert the energy that available from the natural form to the noble forms which can be further used to produce energy. The energy transformation occurs in the types using its important properties by heating and through the flow percentage. The mainly used primary energy machines are,

Steam boilers, wind turbines, and nuclear reactors.

2.1.1 Wind turbines

The wind is the amount of air moving along a certain direction with a certain velocity. It is created by uneven heating of the surface of earth by the sun. This motion of the wind contains some kinetic energy and it is available in nature. This is a renewable form of energy; this can be used to produce electricity by wind turbines. Wind turbines are machines consists of blades mounted to a tower, most of the wind turbines consist of three blades. wind turbines catch the wind’s energy with their propeller like blades.

When the air blows a lower pressure air forms on one side of the blade which pulls the blade towards it causing the rotor to turn. This rotor is connected to a generator that produces the required electricity. Wind turbines have a variety of application form place to place it changes, like from harnessing offshore wind resources to generating electricity for a single home. Figure 1 shows the primary energy transformation takes place in the wind farms.

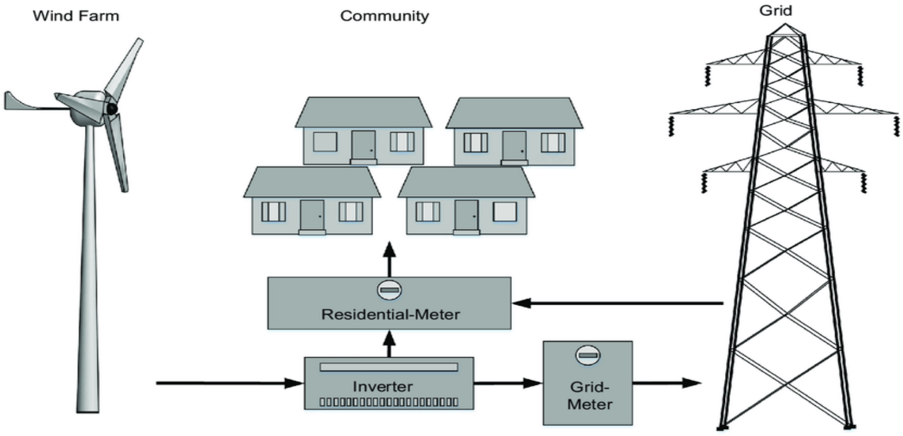


Figure 1 Wind turbine [1]

2.2 Secondary machines

Secondary machines are those which are used to convert one form of energy into another form of energy as did in the primary energy transformation engines, but the difference is that these are not natural energy or natural resources. Like for example turning on a fan converts electrical energy into mechanical energy to produce wind. Some commonly used secondary energy machines are,

Water pump, air compressor, axial steam turbine.

2.2.1 Air compressor

Air compressor is a machine that is capable of converting electrical power to mechanical energy and into kinetic energy by utilizing compressed air. The air itself release kinetic energy if it is burst quickly, where this energy is used for air transfer, pneumatic device activation, and cleaning operation. The compression phase and the emission phase are the two-working phase of an air compressor, some of the compression phase methods include the piston reciprocation, rotary screw, and centrifugal compression. They are measured in cubic feet per minute. Thus, more the horsepower, more is the air delivery from the compressor. Here we see the scheme of air compressor used to generate electricity, as shown in Figure 2.

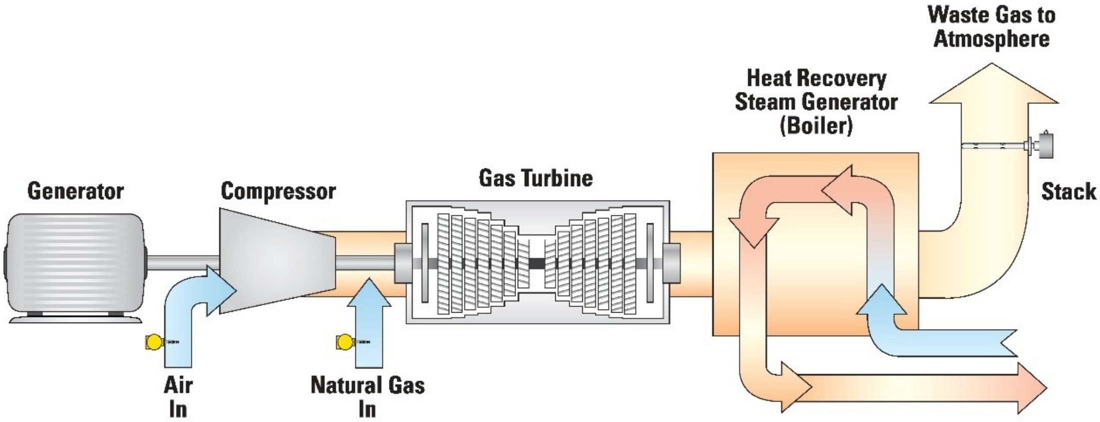


Figure 2 Air compressor in power generation [2]

2.3 Tertiary machines

Tertiary energy machines are the last type of energy transformation machines, where the transformation of energy takes place between the same form of energy medium. So, we also call it "the changers". As this becomes the tertiary machine, it concentrates only on transferring energy mediums such as mechanical, electrical, enthalpy of work (exchange of heat) for various industries and its application. In which some of them are,

Gearbox, Electric Transformer, Heat Exchanger.

2.3.1 Heat exchanger

A heat exchanger is a mechanical device used to transfer heat from one fluid to another fluid or it may be gas. It is used both in the cooling and heating process. Heat exchangers can be used for a variety of purposes and can be used to add or remove heat from the water, oils, or other liquids that it is used with. An example of where heat exchangers are used is in the heating of outdoor pools. In this instance, hot water from a boiler or other water reserve is transferred through pipes underneath the pool to transfer the heat into it. The two main types of heat exchangers are tube heat exchangers and plate heat exchangers. Figure 3 shows the energy transformation in the heat exchanger.

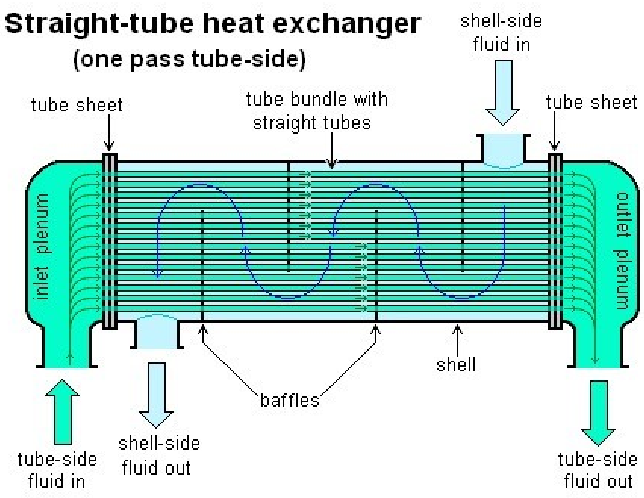


Figure 3 Tube heat exchanger [3]

The energy transformation diagram in Figure 4 shows the exact cycle of energy usage and the machines which use those energy supplies.

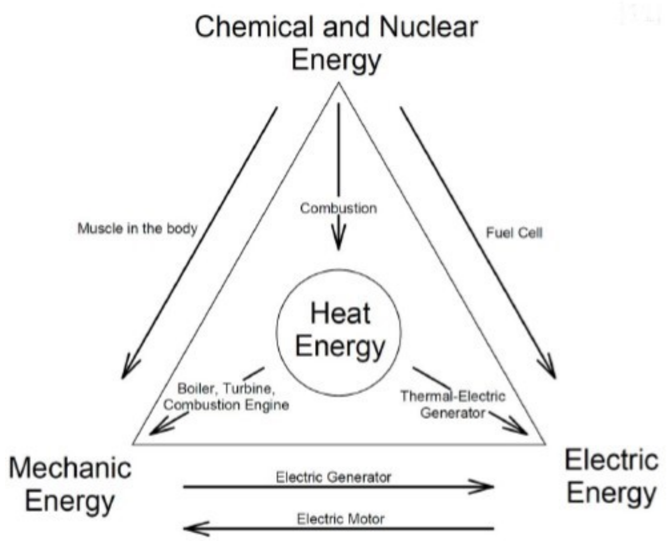


Figure 4 Energy transformation cycle

3 COMPRESSORS

A compressor is a mechanical device that is used to increase the pressure of the gas or the medium that flows through it. The word “compressor” gives us the meaning that the medium pass through it gets compressed, that is the medium is compressed to increase their pressure and reduces its volume. In most common applications the air is used but natural gas, oxygen, and other industrial gases are also compressed. Compressors are also compared to pumps, both increases the pressure of the medium that passes through it, but mostly liquids are incompressible.

As the human lifestyle has changed has a part with compressors by contributing its application in the revolution of the industries in the modern world. The fuel we use for our resources crude oil will not become stabilize without compressors. As not only in that compressors play a lot in our day-to-day lifestyle that we normally won’t realize that’s existence. Compressors are used in jackhammer, rock drills, in the mining process, spray guns, automobile washing even we fill out automobile tire with air by use of compressors. About 10 to 30% of the world's electricity is used to run the compressor.

Let's see some interesting facts about compression that take place naturally that we haven't known that is because of compression. Thunder is the sound created by the air exploding, yes when a lightning bolt strikes the earth the next second it travels back to the sky. This secondary bolt heats the air immediately around 27000°C. This heated air has no time for expansion and becomes pressurized from and produces up to one hundred times more than atmospheric pressure and then it becomes explode and releases the energy in the form of energy.

Likewise, the compressor generates a huge amount of heat energy while producing compressed air, which can be used to provide heating or hot water to the buildings.

There are two ways or principles for the compression of gas or air, the displacement compression, and the dynamic compression. Among the displacement compression, the air is drawn into a compression chamber and then its volume is reduced by compressing that is kind of piston compressor. Were the displacement compressors which is most commonly used in most countries.

The dynamic compression is done by rotating blades or fan that will increase the velocity of the air and then the air is discharged to the outlet with the help of a diffuser, where the kinetic energy of the air changes into static pressure and then the compression is done. Though in dynamic compressors there are two kinds of flow axial and radial compressors. These compressors are used for large volume rate of flow.

3.1 Types of Compressors

The compressors are divided according to the following,

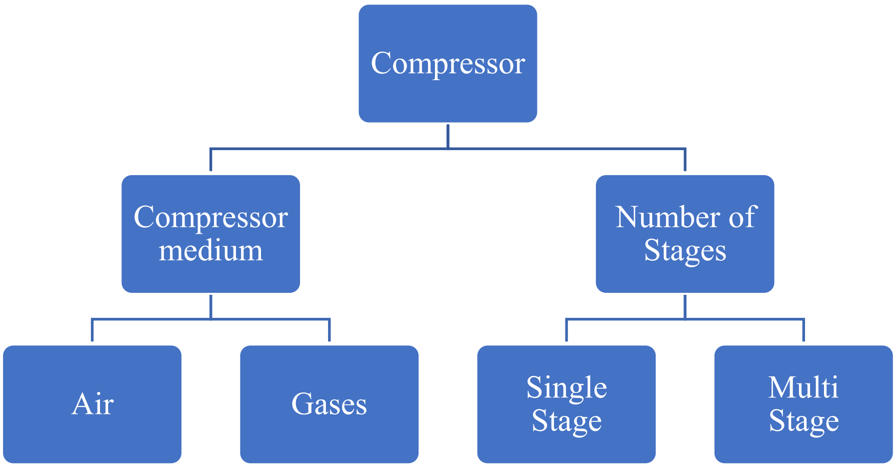


Figure 5 Types of compressor

Including these types, other parameters tend to classify compressors according to the characteristics,

- Based on total pressure ratio σ_c
- Based on compressor flow performance
- Based on the cooling medium
- Based on mobility

3.1.1 Based on total pressure ratio σ_c

The total pressure ratio is defined as the ratio between pressure inlet and outlet of the whole system.

$$\sigma_c = \frac{p_d}{p_{n,l}} \; [-] \qquad (1)$$

Where,

$p_d \; [pa]$ – pressure at piping system

$p_{n,l} \; [pa]$ – the pressure of the suction gas

Based on the total pressure ratio the compressors are classified into the following

- Blowers $\sigma_c < 3$
- Low pressure compressors $3 < \sigma_c < 25$
- Middle pressure compressors $25 < \sigma_c < 100$
- High pressure compressors $100 < \sigma_c < 300$
- Hyper compressors $\sigma_c > 300$

3.1.2 Based on compressor flow performance

The volume of the flow rate of gas which is carried over through the outlet pipe of the compression system with the help of the suction condition is called the compressor flow performance. According to the law of conservation of mass, the transported amount in the mass flow is given as,

$$\dot{V}_d = \frac{\dot{m}_d}{\rho_s}\ [m^3.s^{-1}] \tag{2}$$

Where,

$\dot{m}_d\ [kg * s^{-1}]$ – transported amount

$\rho_s\ [kg.m^{-3}]$ – density of gas in the compressor suction pipe

Based on maximal compressor flow performance it is divided as,

- Small compressors $\dot{V}_d < 150\ m^3.h^{-1}$
- Middle compressors $150 < \dot{V}_d < 5000\ m^3.h^{-1}$
- Large compressors $\dot{V}_d > 5000 m^3.h^{-1}$

3.1.3 Based on the cooling medium

The cooling medium is used in a compressor to prevent overheating or adiabatic heating, based on the amount of heat generated in the compressor the cooling medium is used in different methods. The types of cooling medium used are,

- Air cooling method
- Water cooling method
- Oil cooling method

Air cooling method is used for suction air returning from the compressor or supplying air throughout the compressor and this type of cooling is commonly used in semi-hermetic compressors

The second method of cooling is by using water as a coolant around the compressor body and these types of compressors are generally called water-cooled compressors.

Mostly screw compressors use oil as a cooling medium here the oil seals, cools, and then it lubricates the compressor and these type compressors are commonly called oil-cooled compressors.

3.1.4 Based on mobility

Compressors are used in all different sizes and at different characteristics, among those some need to be carried from one place to another for personnel requirements. Based on its size of the compressor it is commonly divided into three types,

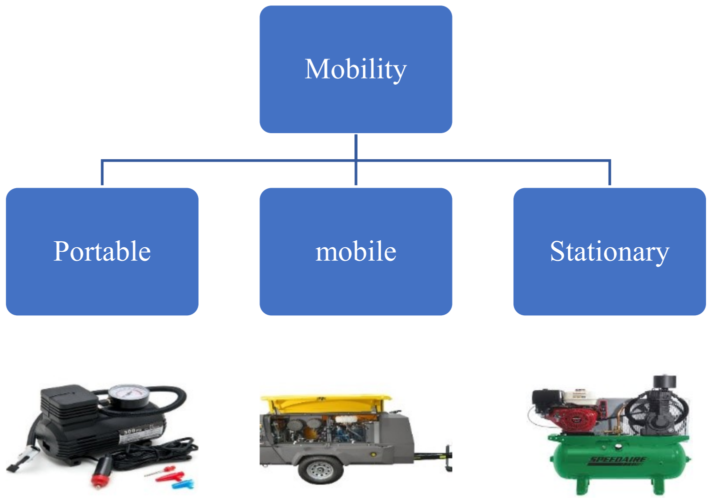


Figure 6 Types based on mobility [13][14]

Given the good size of the portable compressor, it cannot be used in all situations, i.e. where high pressure compressed air is required. In this scenario using the handheld compressors and the stationary compressors. The portable compressors are used in our daily routine to inflate motorbike and car tires. Focused on the compressor compression process they are primarily classified into two categories as,

- Displacement compressors
- Dynamic compressors

3.2 Displacement compressors

A displacement compressor or positive displacement compressor is used to compress natural gases that can be further transported to industrial processes. In displacement compressors, the pressure energy increases with a decrease in the volume of the workspace. The compressors are secondary energy transformation engines that convert one form of energy into another form, i.e. the electric energy is converted into pressure energy. In a displacement compressor, the pressure energy is increased by decreasing the volume of the gas. Displacement compressors are used to achieve higher temperatures and only for smaller amounts of gas.

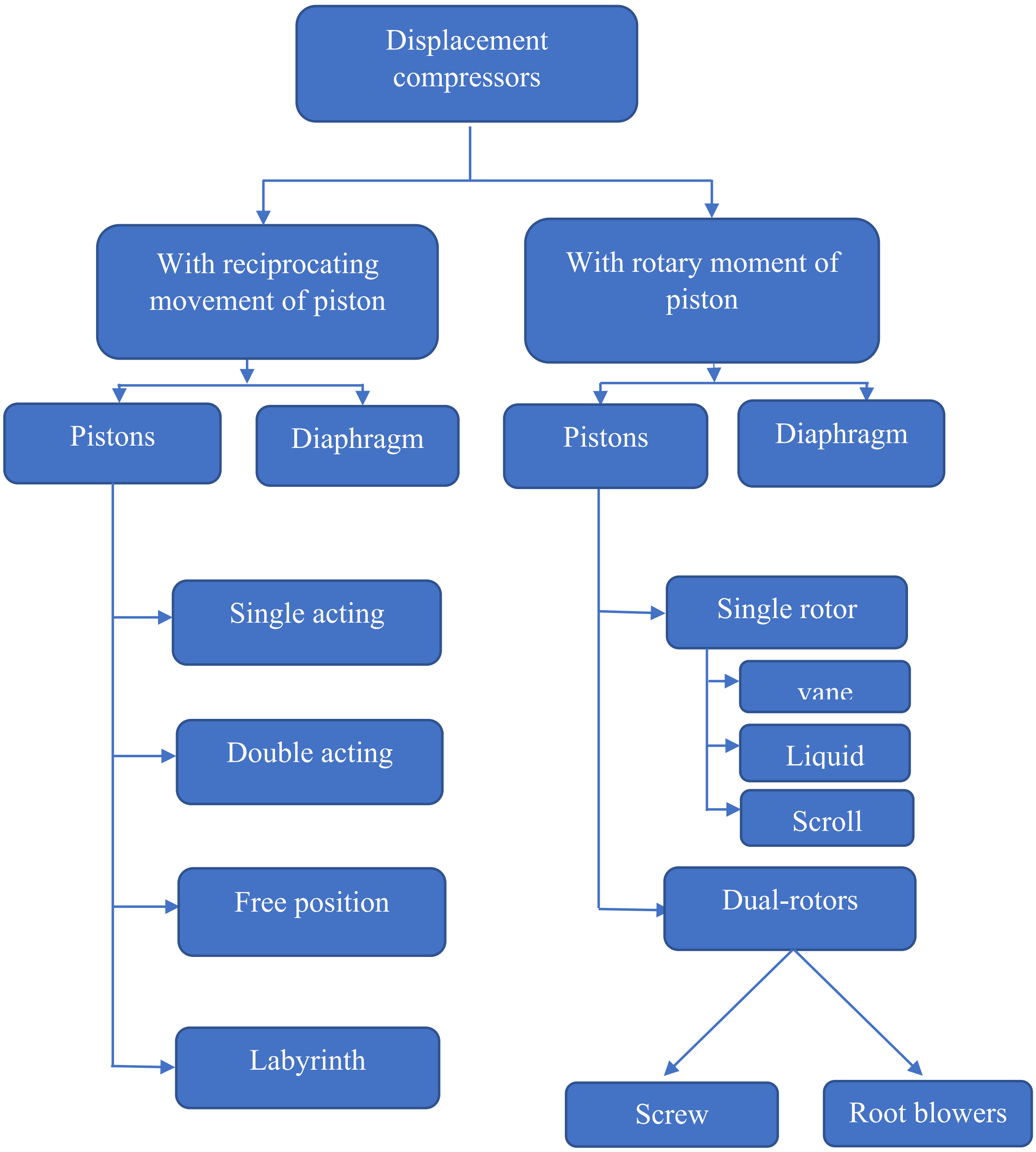


Figure 7 Types of displacement compressor

3.2.1 Features of Displacement compressor

- In displacement compressor occurs the pressure energy increases (to the compression process) due to a volume decrease of the working space.
- The compressors are the secondary energy machines, in which occur the transformation of the “noble” energy form (Eel, h) to the pressure energy of the working fluid.
- These are machines designed for the compression and the transport air and other technical gases.
- When we compare the displacement compressors with the dynamic compressors – the displacement compressors are used for higher pressures and the transport of the lower amount of gas.

4 DYNAMIC COMPRESSORS

Dynamic compressors are rotary continuous flow machines in which the rapidly rotating element accelerates the air as it passes through the element, converting velocity head into pressure. Where the pressure energy increases due to the change in momentum of the gas flow in the working space. The difference between the dynamic compressor and displacement compressor is that the displacement compressor works with constant flow and the dynamic compressor works with constant pressure. Dynamic compressors are available in axial and radial designs. The latter is frequently called turbo or radial turbo and the former are called centrifugal compressors. The capacity of a dynamic compressor varies with the working pressure. The characteristics of dynamic compressors are given below [11].

- The compressors are the secondary energy machines, in which occurs the transformation of the "noble" energy form to the pressure energy of the working fluid.
- These are designed for the compression and transportation of air and other technical gases.
- In a dynamic compressor, the compression process occurs when the working medium contacts with the outside environment on the outlet of the working space.
- The dynamic compressors are used for the low pressure and the transport of a higher amount of gas.

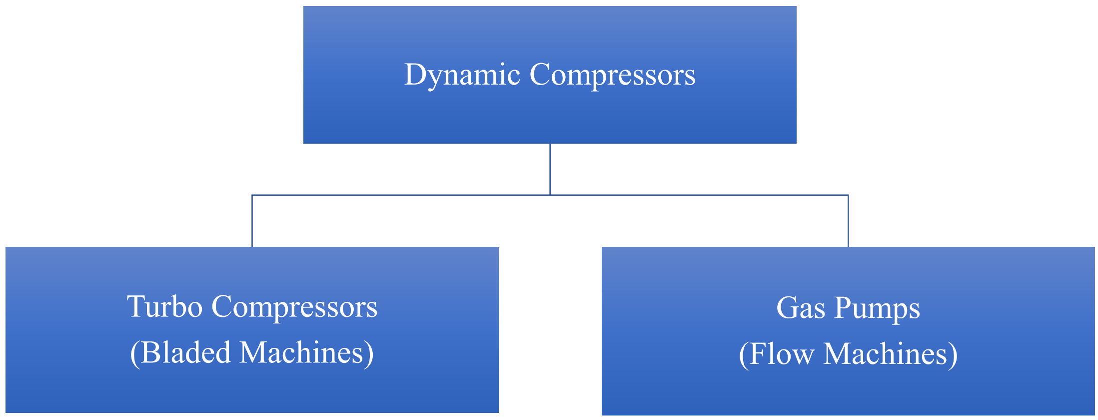


Figure 8 Types of dynamic compressor

4.1 Flow machines

Flow machines are devices that are used to convert the pressure energy of the moving fluid or gas into velocity energy by using the help of nozzles. They do not have any moving parts and are suitable for high flow performance with a simple design and cheaper manufacturing. These machines are used in,

- Extraction of gases, liquids, and vapor.
- Mixing or dilution of chemicals like acids or bases.
- Stirring.
- Increasing pump suction pressure.

But these flow machines are expensive for their designs, for different applications we require different computational models and are sensitive to changing operating conditions.

A nozzle is a machine component with a continuous change in the flow cross-section. The working of the nozzle is, when an amount of gas is passed through a flow cross-section there occurs a decrease of pressure and temperature and occurs the increase of the velocity of the flowing fluid. Which is simply said as a decrease in the enthalpy and the increase of the kinetic energy. Generally, nozzles are divided into two the conical nozzle and the convergent-divergent nozzle.

A **conical or convergent nozzle** is a nozzle with a larger cross-section at the first and decreases in cross-section along the length of the compressor. The machine part has the shape of a confuser. To maintain the constant amount of flow medium through the restricted portion of the nozzle, the medium must move faster. The increase in velocity and the decrease in pressure are shown below in Figure 9.

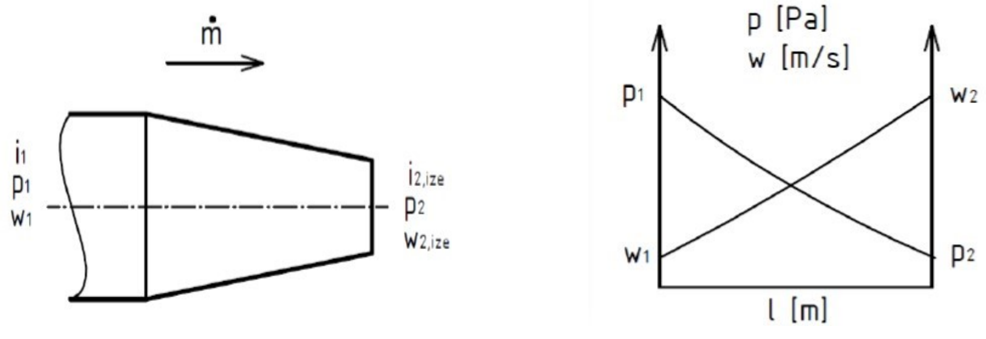


Figure 9 Conical nozzle

The **Laval or the convergent-divergent nozzle**, like the name, mentioned the nozzle has confusion on the input side (with decreasing cross-section) and the output side with a diffuser (with increasing cross-section). This kind of nozzle is used to improve the efficiency of gas expansion beyond the critical cross-section, which is in the case of critical or supercritical flow.

Therefore, in confuser, the flow is always subcritical were the Mach number is less than one. The next part is the narrowest point achieves the flow velocity the local speed of sound with Mach number equal to one. The final stage is the diffuser part was the supercritical flow occurs and here the Mach number is greater than one. The kinetic energy of the gas at the outlet is greater than the kinetic energy at the critical point because of the supersonic flow in the divergent part. This type of nozzle is mainly used for supersonic aerodynamic tunnels. The decrease in pressure and the increase in velocity of the nozzle is shown in Figure 10.

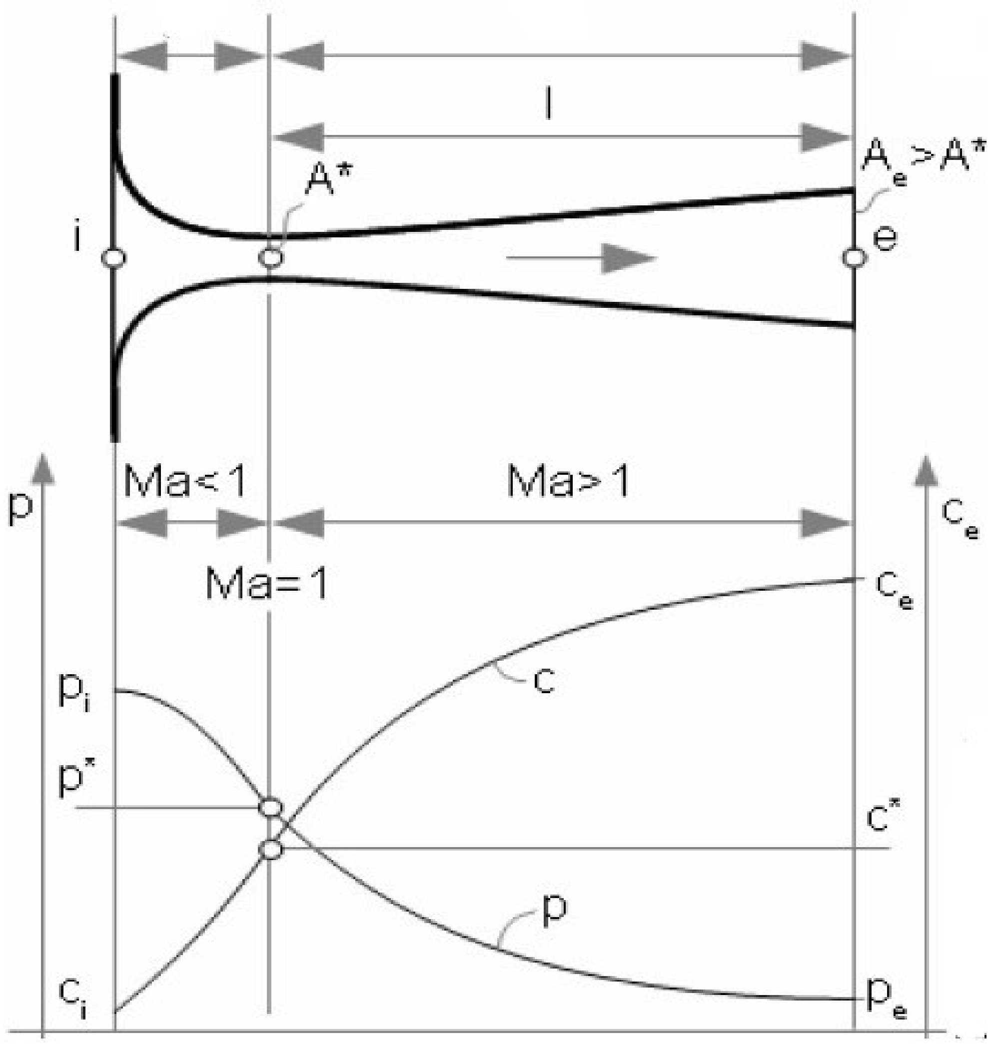


Figure 10 Laval nozzle

Flow machines can be divided into two based on the type of application:

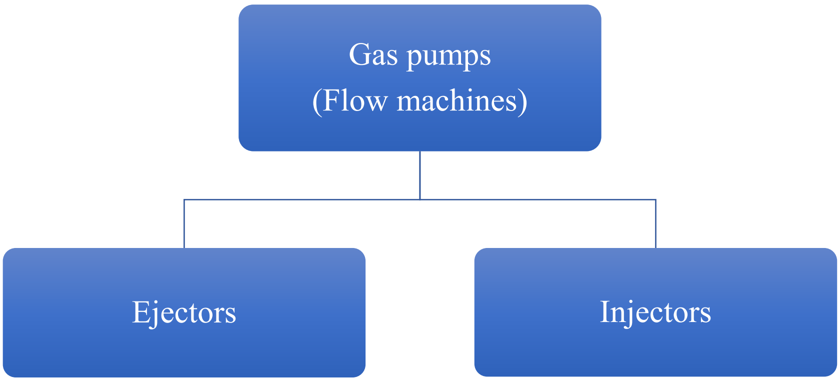


Figure 11 Types of flow machines

4.1.1 Ejectors

This device is used for the extraction or suction process. These are very common to Vacuum pumps or compressors, with the difference that ejectors have no moving parts. At the outlet of the ejector, the pressure is lower than the inlet pressure of the primary medium. The operating principle of the ejector is the pressure energy of the fluid is converted to velocity energy with the help of Laval (convergent-divergent nozzle). Figure 12 is the diagram for the ejector with the direction of flow.

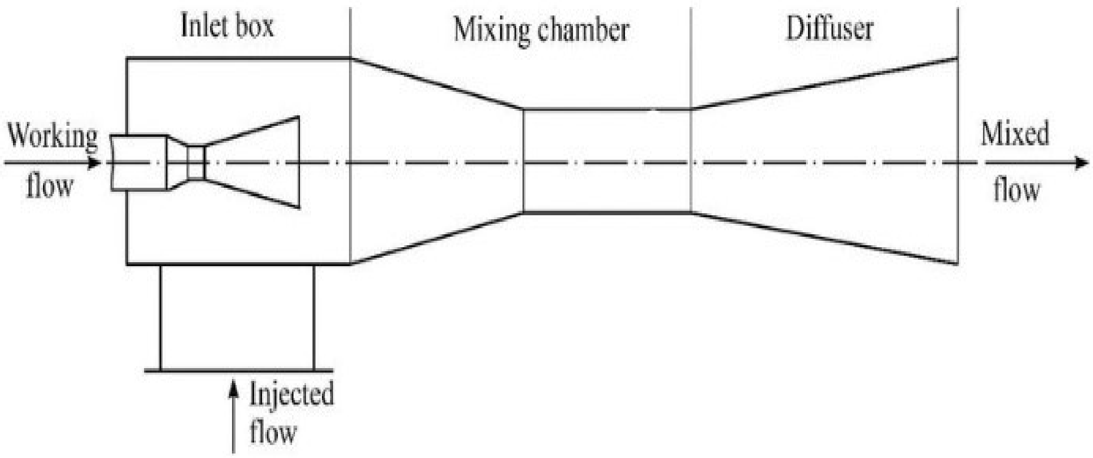


Figure 12 Ejector [4]

4.1.2 Injectors

These are opposite to the ejectors; these devices are used for increasing the pressure energy. At the outlet of the injector, the pressure is higher than the inlet pressure of the primary medium. These injectors play a major role in automobile industries, where the fuel injection is carried out with injectors. The picturized view of the injector is shown in Figure 13.

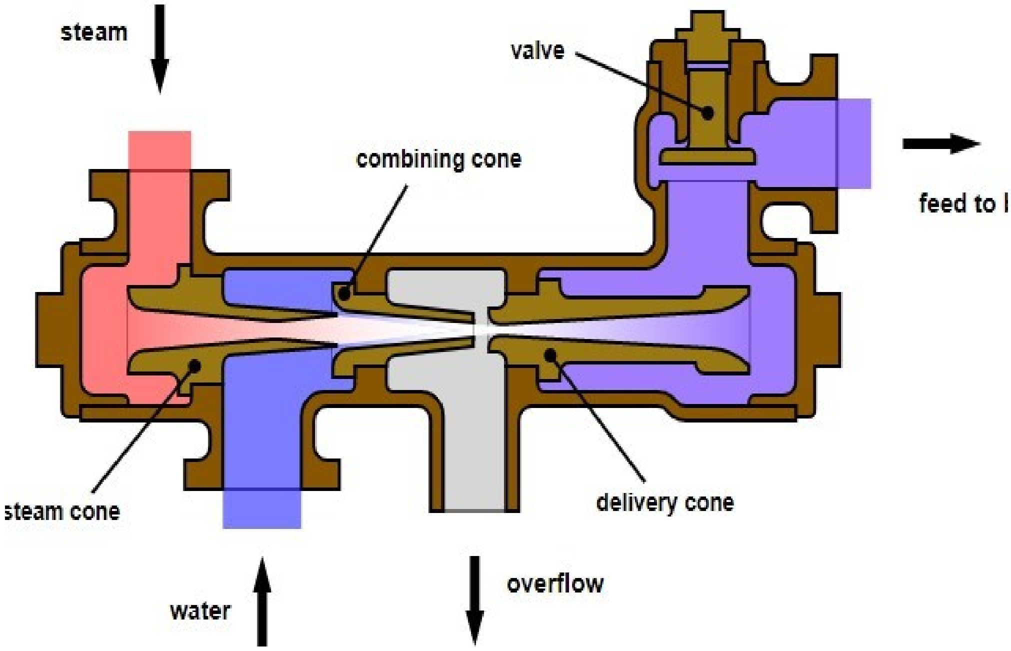


Figure 13 Injector [5]

4.2 Turbo compressors

Also called bladed machines, these are secondary energy machines in which the transformation of noble energy to pressure energy of the working medium occurs. Bladed machines are used for compression and the transportation of air and other technical gases. Here in this process of compression, the pressure energy increases due to the change in momentum of the gas flow in the working space. Dynamic compressors are in areas where low pressure and higher amount of air transportation is required. If we are looking for a compressor with huge horsepower, then a dynamic compressor is the ideal choice.

The bladed machines are divided into,

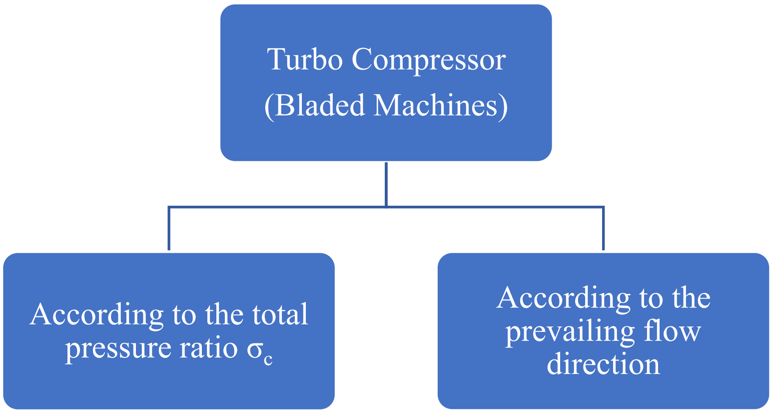


Figure 14 Types of turbo compressors

4.2.1 According to the total pressure ratio σ_c

According to the total pressure ratio, the bladed machines are divided into three divisions. They are,

- Fans (from 1.01 to 1.1)
- Blowers (from 1.1 to 3.0)
- Compressors (greater than 3)

4.2.1.1 Fans

They are secondary energy machines that serve to transport a large number of gases for industrial purposes and have a low pressure i.e., from 1.01 - 1.1. this can be applied because there is no increase in the internal energy of the working medium respectively and its temperature would not increase. The blades of fans are bent in the direction of rotation. Here the low direction is parallel to the axis of rotation of the machine and is mainly single-stage machines (part of micro-cooling towers). At the inlet, to the rotor, it is mounted the spherical cap to reduce the aerodynamic damage caused by air. The fan can be designed with counter-rotating rotors and are called “Reverse fans”.

Even fans are divided into,

- According to the pressure ratio
  - Low pressure $\sigma = (1.01 \div 1.03)$ [-]
  - Middle pressure $\sigma = (1.03 \div 1.06)$ [-]
  - High pressure $\sigma = (1.06 \div 1.1)$ [-]

- According to the prevailing flow direction
  - Axial
  - Radial
  - diagonal

Blades bent in the direction of rotation transforms more added work into kinetic energy.

$$1 < \tau < 2, 0 < K < ½$$

Where τ is the ratio of the peripheral component of the absolute velocity to the peripheral velocity called blade shape factor. k is the reaction factor – which describes what portion of the externally applied function converts in the pressure energy in the rotor blades.

In the **axial flow fan**, the air blows forward, which moves in the axis of the fan without centrifugal effect. Generally, an axial flow fan is suitable for a large flow rate with relatively small pressure gain and centrifugal fan for a comparatively smaller flow rate and a large pressure rise. Depending on the purpose and the quality of air handled the size and materials the fans are decided. They are used to supply fresh air, exhaust air out, etc. the axial flow fans are classified into:

- Propeller
- Tube axial
- Vane axial

Centrifugal fans are a mechanical device used for moving air or other gases in a direction at an angle to the incoming fluid. Radial fans are industrial workhorses because of their high static pressure and ability to handle heavily contaminated airstreams. These radial fans are well suited for high temperature and medium blade tip speeds because of their simple design. The working of the fan is described by, the fan increases the kinetic energy of the impeller to increase the volume of the air stream, which in turn moves against the resistance caused by ducts, dampers, and other components[18]. The blades according to their classification is shown in Figure 15. The centrifugal fans are classified into types base on their requirements:

- Backward swept blades
- Radial blades
- Forward swept blades

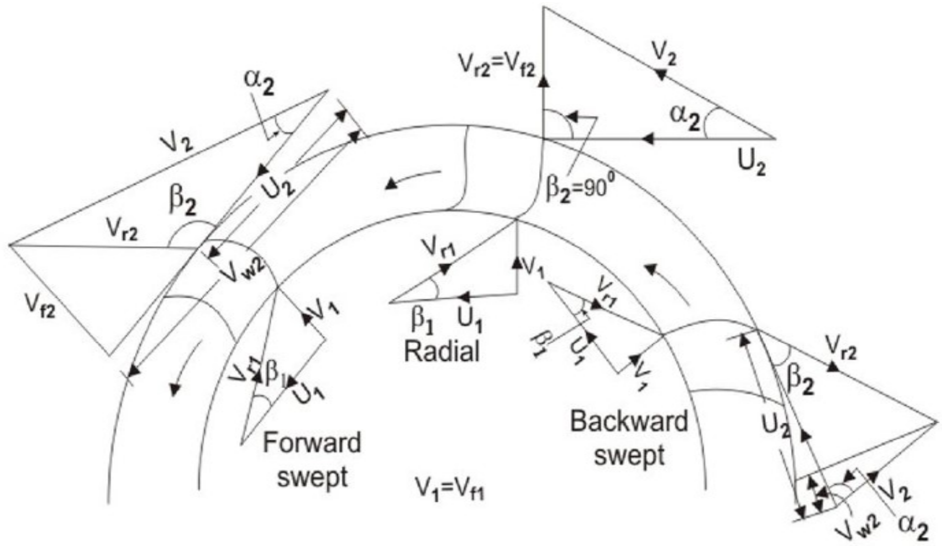


Figure 15 Velocity triangle for different blade shapes [6]

4.2.1.1.1 Backward swept blades

The centrifugal fan is named after its direction of flow and how the working medium exits the impeller (example air) radially from the outer circumference of the fan. A backward swept fan blade is distinguished by its cylindrical form, multiple large curved blades, and a conical nozzle. A pressure difference is created on the impeller when the fan starts rotating. The fan now generates a positive pressure from the convex side of the impeller blade imparts a force in the air by the rotary motion. The blade pushes the air in a radial direction, exiting. A negative pressure is created on the concave side of the impeller blade, as the fan rotates, pulling air into the space between the blades. This air is then raised by the next blade and radially forced into a continuous process. The suction side of the rotor blade draws air from the center of the ventilator, which caused a directional change in the airflow from the inlet to the 90°exhaust. Figure 16 shows the backward swept blade [15].

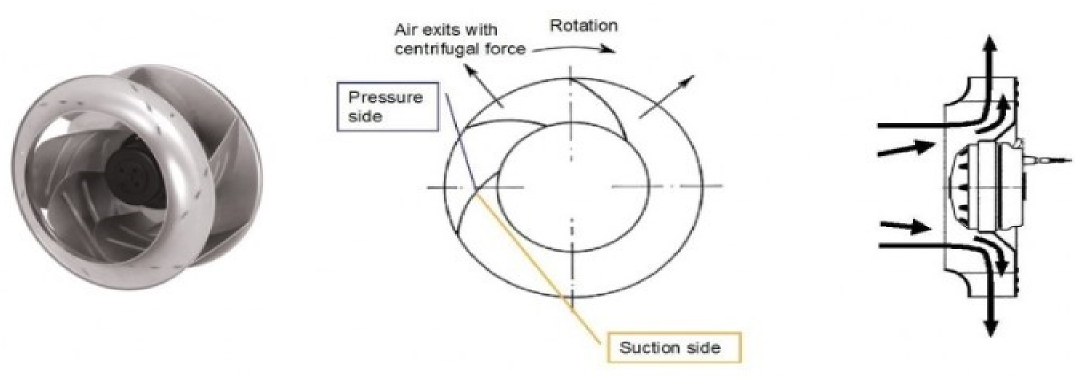


Figure 16 Backward swept blade [15]

4.2.1.1.2 Forward swept blades

The centrifugal fan is named after its direction of flow and how the working medium exits the impeller (example air) radially from the outer circumference of the fan. The difference between the backward swept blade and the forward-swept blade is the direction of air flows from the impeller. The air leaves the fan in a radial direction with a backward curved impeller while the air leaves the fan tangentially with a forward curve. It is characterized by its cylindrical shape and several small blades on the circumference of the impeller. For forward-curved blades, it is necessary to provide housing that converts high-velocity air into a lower velocity static force[18][20]. The housing shape also influences the airflow at the outlet, is shown in Figure 17 which is also commonly known as a scroll or volute.

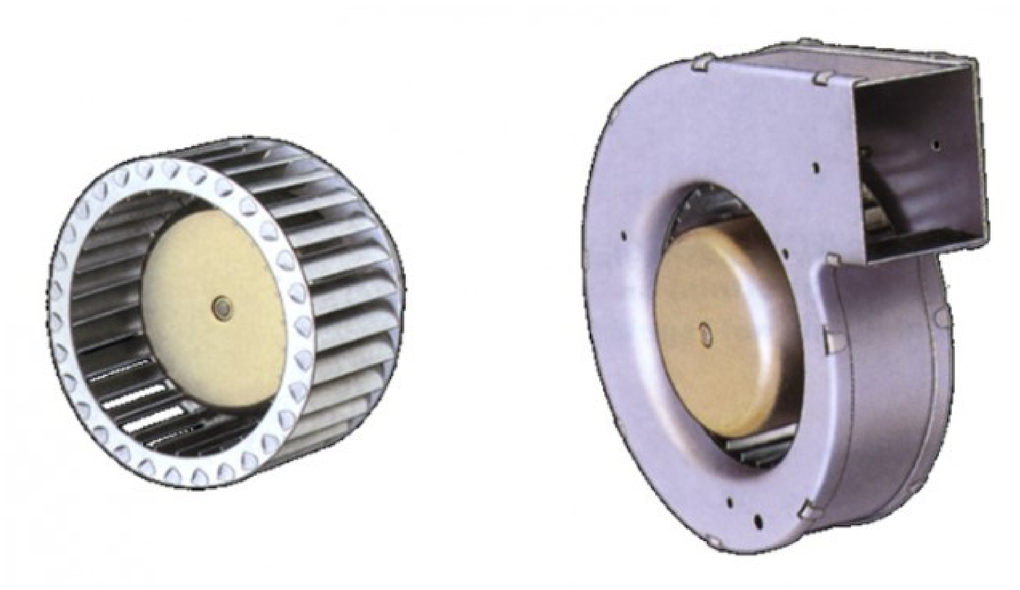


Figure 17 Forward swept blade [20]

4.2.1.1.3 Radial bladed fans

Radial fans also called a “steel plate” or “paddlewheel” are commonly used to move gases and other process material within the industry. The blades of the impeller are normally shorter, narrower, and smoother than the forward curved and backward inclined blades. These are usually high-pressure devices that use combustion air to manage the air and perform well in robust environments. The best structural architecture in the field is also used for radial blade fans. They are robust, versatile, and heavy enough to transfer vast volumes of air – allows virtually any industrial fan design to reach the highest pressures.

Radial blade fans may be retained in addition to being strong. Radial blades will save money over the long term due to how simple it is for systems concerns with high constant maintenance costs – especially for certain more sophisticated setups. As shown in Figure 18.

The self-cleaning properties which most radial blade fan designs have are part of the reason they are simple to maintain. Mechanisms of self-cleaning are important as they reduce labour expenses and help enhance organizational life [18].

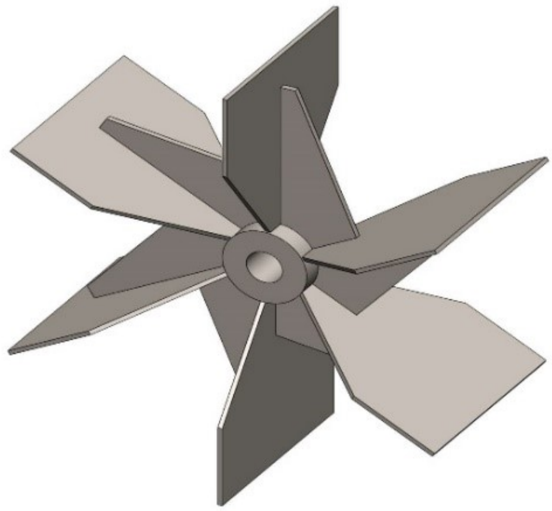


Figure 18 Radial blade [21]

4.2.1.2 Blowers

Blowers these secondary energy transformation device increases the velocity of air when it is passed through the impeller. Blowers are mainly used in the flow of air or gas required for ventilating, cooling, exhausting, aspirating, conveying, etc. In blowers, the working medium experience, a low pressure at the inlet and is higher at the outlet. This is because the kinetic energy of the blades at the inlet increases the potential energy of the air at the outlet. Blowers are mainly used in industries where moderate pressure is required, that is the pressure is more than the fan and lower than the compressor. The blower is shown in Figure 19. Blowers are divided into two,

- Centrifugal blowers – like centrifugal fan but gains higher pressure.
- Positive displacement blowers – the constant volume of air.

Figure 19 Blowers [7]

4.2.2 According to the prevailing flow direction

"The flow of the transported and compressed gas occurs predominantly along surfaces perpendicular to the axis of rotation of the machine". According to the prevailing flow direction, the bladed machines are divided into three divisions. They are,

- Axial flow compressors
- Diagonal or Mixed flow compressors
- Radial flow compressors

4.2.3 Axial flow compressor

"Rotary bladed secondary energy transformation machines where the transported and compressed gas flows along cylindrical surfaces, that is parallel to the axis of rotation of the machine". The air or gas passes through the compressor shaft through stages of rotating and stationary impellers. In this way, the velocity of the air is increased gradually and at the same time, the stator converts the kinetic energy of the air to pressure energy. Figure 20 shows the axial flow compressor.

Axial compressors are generally smaller than the centrifugal or radial compressors and work with about 25% higher speed ordinarily. These compressors are used for a constant high-volume rate of flow at relatively moderate pressure. The pressure ratio of the axial compressor is higher than 6, and the normal flow is approximately 65 m^3/s and pressure approximately 14 bar. However, these machines have been developed to design compressors with higher efficiency, and investment cost for these machines at high flow performance is lower than the radial compressor by up to 30% [11]. The Characteristics of the axial compressor are given below.

- Energy efficiency is from 2 to 5% higher
- They are suitable for pressure up to 1.5 Mpa
- The flow performance from 10000 to 2500000 $m^3.h^{-1}$
- The speed from 3000 to 20000 RPM
- The compression curve is the same for both RTC and ATC

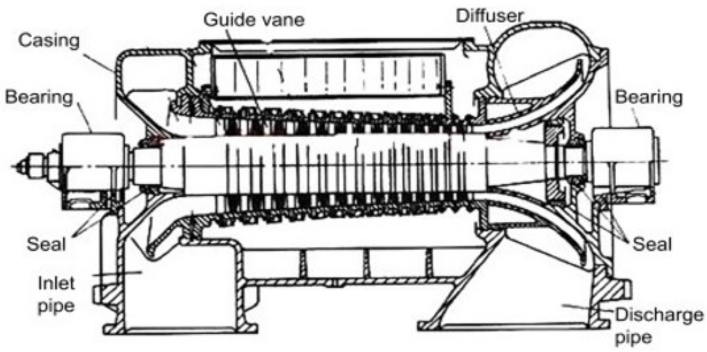


Figure 20 Axial compressor [8]

4.2.4 Centrifugal Compressors

Thus, in centrifugal compressors (also known as a radial compressors), energy transformation takes place by air drawn into the center of the rotating impeller with radial blades and is pushed out towards the perimeter of the impeller by centrifugal forces. This accelerates the working medium and increases its kinetic energy. Before the air is led to the center of the impeller of the next chamber, it is passed through a diffuser and a volute where kinetic energy is converted into pressure energy or potential energy. In the radial stage, the flow occurs mainly occurs in the radial direction that is, the flow is perpendicular to the axis of rotation. As centrifugal compressors are capable of produce a higher-pressure ratio at lower flow rates than axial compressors can produce. Each stage of the impeller increases the pressure, according to the pressure ratio required the number of stages can be increased in series [9].

There are two types of flow inward flow and outward flow type. Where gas compressors mainly use inward flow type. The outward flow types are chosen to accommodate a higher volume flow rate of rapidly expanding steam. Radial flow stages are not suitable for high flow rates, because in radial flow the gas flow turns into 90° traversing. So, a much longer blades compared to axial types, which leads to higher losses and also lower efficiencies.

Like in multiple stages the flow is required to change its direction several times drastically which affects the design mechanically and aerodynamically. Therefore because of this reason, most of the radial machines are single-stage machines. Though using the radial machines at a single stage is more rigid and reliable compared to axial stage machines.

On account of higher peripheral speed and additional change of energy level in gas caused by centrifugal energy, much higher values of the pressure ratio per stage are obtained in the radial stage compared to axial types. These radial types are not used for aircraft because of its large frontal area. The difference between the centrifugal compressor and centrifugal fan is the pressure produced and the number of stages of the impeller [12]. Let's see the list of main parts of the centrifugal fan in Figure 21.

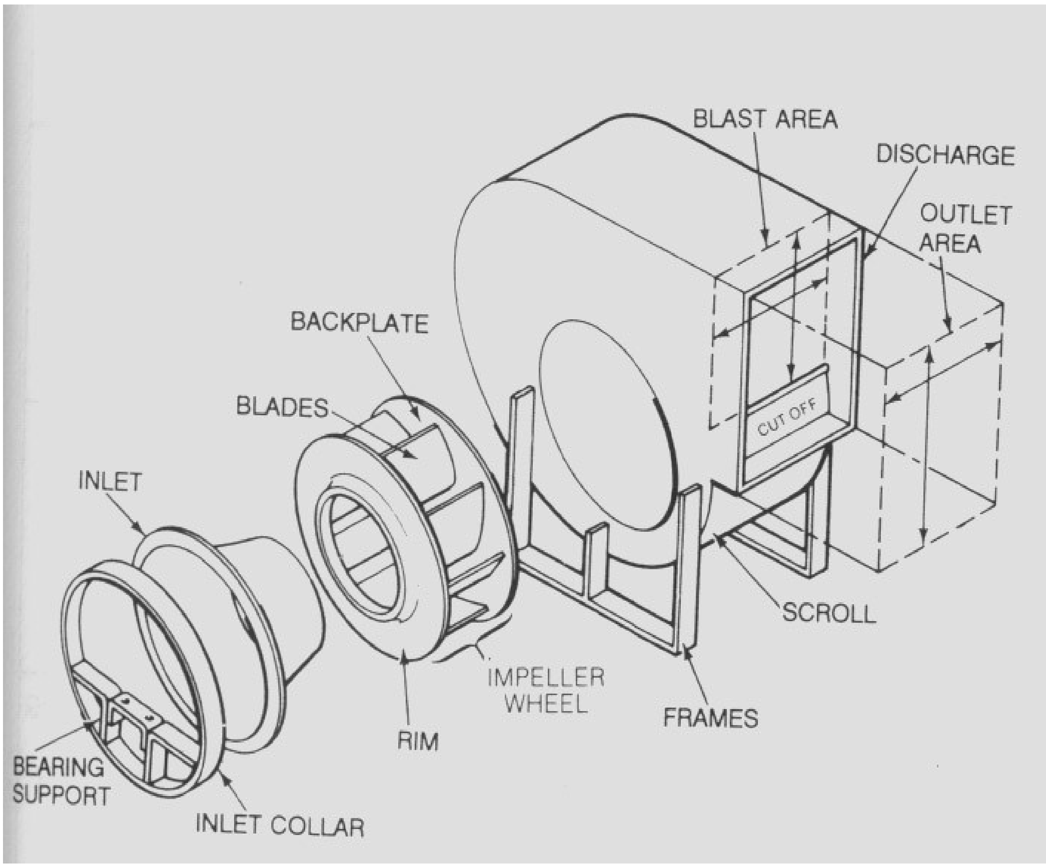


Figure 21 Parts of centrifugal fan [12]

4.2.4.1 Inlet

Inlet or inlet cone is a stationary part in the working process of centrifugal fan, is fixed with the casing, and directed towards the center of the fan. The purpose of the inlet cone is to convey the flow of gas to the fan wheel. Without inlet cone, there may be difficulty in achieving a constant path for airflow. Placing the inlet cone in the wrong position causes loss of airflow.

4.2.4.2 Impeller wheel

An impeller is considered as most important in the design of a centrifugal fan or compressor. It has three parts like backplate, rim, and blades combinedly called the impeller wheel. The backplate holds the blade and the rim. The blades are made in different shapes according to the working medium, environment, and the pressure required. The rim is the closing part of the impeller wheel which supports the blade design to increase the pressure energy. These impeller wheels are connected to the shaft from the motor.

The working process of the radial impeller wheel, when the shaft starts to rotate the is a suction of air is generated due to the radial shape of the impeller. The airflow is regulated with the help of an inlet cone. The air inside the impeller increases the pressure and the velocity of the air. The radial impeller produces an output at right angles to the impeller.

4.2.4.3 Volute

Volute, which is known by scroll, housing, etc. volute is the outermost cover of the radial compressor with increases in cross-section. The working medium after leaving the impeller reaches the volute, it collects the flow by redirecting it in a circumferential direction and delivers it to the compressor or fan outlet. During the flow of working medium in the volute experience an increase in static pressure (potential energy) and decrease its velocity (kinetic energy). The volute cross-section is designed is different shapes according to the requirement.

The other parts including bearing supports, fasteners, flanges, frames are supporting parts that can be modified, replaced according to the surroundings, and the technical requirements of the compressors.

4.2.5 Characteristics of a radial turbo-compressor

- High circumferential velocity from 110 to 380 $m.s^{-1}$
- High speed from 3000 to 80000 RPM.
- The flow performance from 1000 to 100000 $m^3.h^{-1}$
- The total pressure ratio σ_c is a maximum of 20 not more than 80.
- The ideal pressure ratio is $\sigma = 1.25$
- The ideal specific energy consumption, $c = 0.125$ [$kWh.m^{-3}$]

4.2.6 Isentropic compression

An isentropic process is a thermodynamic process, in which the entropy of the fluid or gas remains constant. This means the isentropic process is a special type of adiabatic process in which there is no transfer of heat or matter occurs. But here we will not use the term adiabatic compression because the adiabatic state can change calculation with the internal friction of the working medium (lower degree of idealization).

The isentropic process can be expressed with ideal gas law as,

$$p.v^k = constant \tag{3}$$

Where there is no heat transfer occurs in the environment

$$dq = 0 \tag{4}$$

$$q = constant \tag{5}$$

Because of the isentropic compression on the next stage, at higher gas temperatures, we would need more compression work than needed in isothermal compression. Where the isothermal compression requires a lower amount of compression work which is shown in Figure 22. The device is used to decrease the volume and increasing the friction in the isothermal compression of a gas. Running on the gas increases internal strength and the temperature continues to rise. The machine must exit the heat and join the atmosphere to retain steady temperature energy.

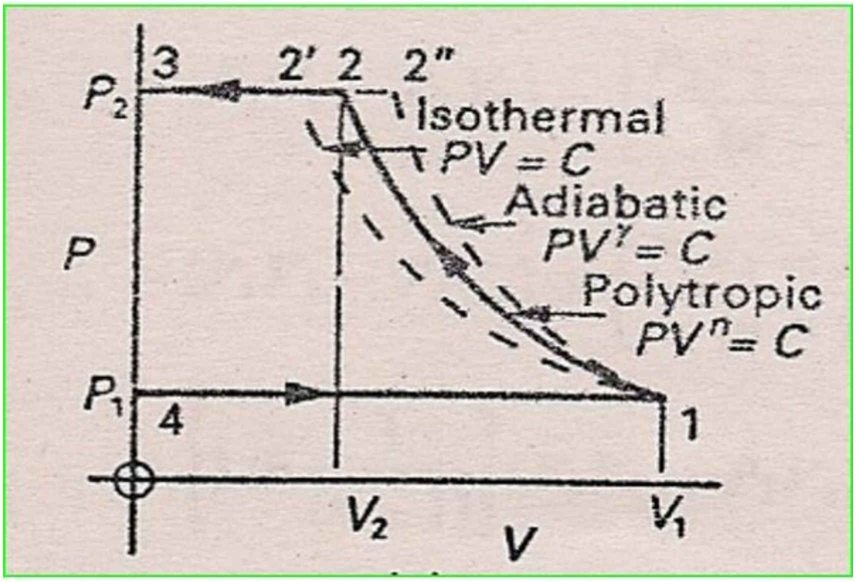


Figure 22 p-v diagram [10]

Here the work done for the compression process in isentropic compression is greater than the work required for compression in the isothermal compression process.

$$a_{isentropic} > a_{iso\ thermal} \tag{6}$$

Sankey's diagram of the energy flow in turbo compressor, Sankey’s diagram is a kind of flow diagram, visualizes the flow quantity proportional to its width. The additional energy added is mentioned by nodes. Let's see Sankey's diagram of the energy flow in the turbo compressor. The energy flow in the turbo compressor is shown in Figure 23.

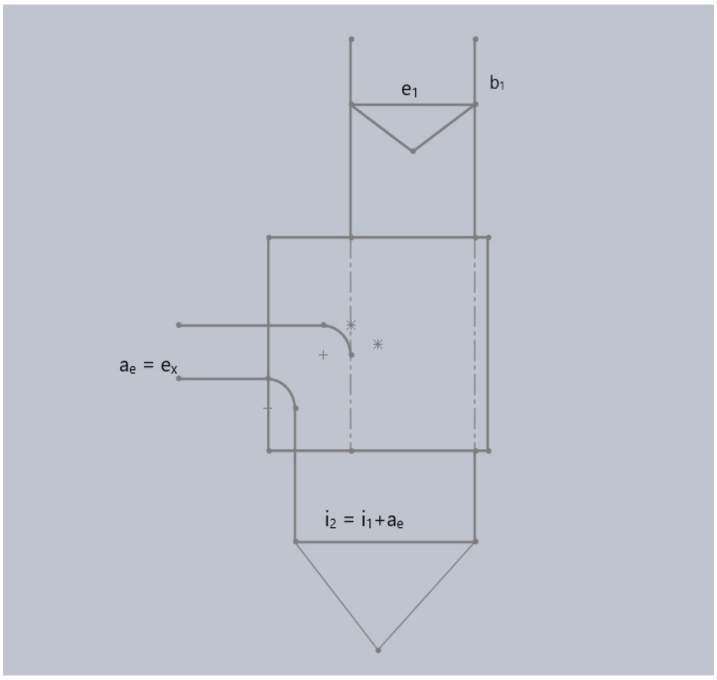


Figure 23 Sankey’s diagram for energy flow in turbo compressor

$$e = b_1 + e_x \tag{7}$$

Where

- Exergy E_x is the part of the energy that can be transformed 100%.
- Energy b_1 is the part of the energy that cannot be transformed (enthalpy of the environment).

Here adding the technical work also increases the enthalpy of the system. While the volume work is necessary for compression work itself and is a consequence of changing the internal energy of the system. The increase in the exergy of the gas after the isentropic compression is equal to the adding of the technical work.

$$a_c = i_2 - i_1 \tag{8}$$

$$a_c = a_{ie} = \frac{k}{k-1} * r * T_1 * [\sigma^{\frac{k-1}{k}} - 1] \tag{9}$$

Where constant specific heat in the isentropic process is given by

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} \quad (10)$$

4.2.7 Stages of the ideal compressor

A compressor stage is defined as one impeller, an inlet guide vane, and the diffuser and namely the seals. Each compressor stage at a given flow and impeller speed will produce a certain amount of energy and have a specific stage efficiency. It is observed that the turbo compressor has the characteristics of producing increase energy only at lower fluid flow assuming the inlet speed and inlet gas angles are constant [24]. The velocities at inlet and outlet are detailed in Figure 24.

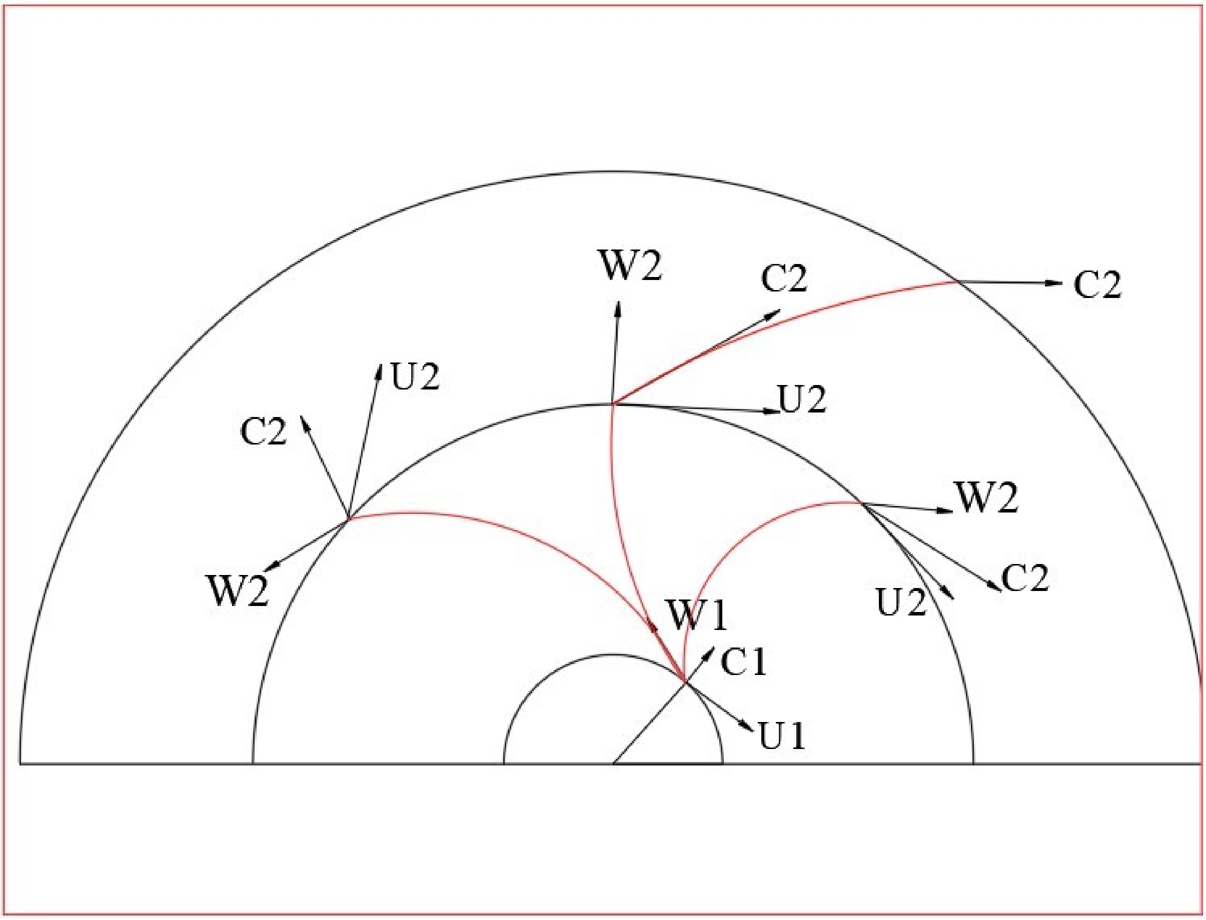


Figure 24 Stages of ideal compressors

The gas enters between the blade space with absolute velocity c_1 is approximately from 20 to 30m.s^{-1}. But there is a peripheral velocity at the rotor inlet has thanks to the rotation process the value u_1. Which in case results in the decomposition of both absolute and circumferential velocity causes relative velocity w_1 which acts tangential to the inlet edge of the rotor blade.

Because of the gas in the rotor moves in the relative environment, the instantaneous relative velocity “w” is always tangent to the rotor blade at given points. The space between the blades of the rotor has the shape of the diffuser where the compression process occurs (a channel with expanding flow cross-section). That is the increase in pressure energy and a decrease in kinetic energy.

Where:

$w_2 < w_1$ decrease in relative velocity in the rotor indicates there is an increase in pressure energy or potential energy.

$c_1 < c_2$ due to adding of work in the rotor the absolute velocity increases.

The continuous process is carried out in the stator part of the compressor. The peripheral velocity at the rotor outlet has thanks to the rotation process of the value u_2. It results in the vector decomposition of the velocities w_2 and u_2 to the absolute velocity at the outlet of the rotor c_2 which is the same as the absolute velocity at the inlet of the stator c_{1s}. likewise, the absolute velocity is tangential to the inlet edge of the stator blade.

The gas moving through the stator in an absolute environment, when the instantaneous absolute velocity “c” is always tangent to the stator blade at a given point.

The space between the blades of the stator has the shape of a diffuser too in which the compression process occurs too

Where:

$c_{2s} << c_{1s}$ there is a significant decrease in the absolute velocity in the stator represents a significant increase in pressure energy or the potential energy.

There is no relative velocity “w” in the stator because the stator is the stationary part of the radial compressor. The diameter of the rotor increases along the gas flow direction where $D_2 > D_1$ due to this the inequality of the peripheral velocity occurs $u_2 > u_1$. Where

$$u = \pi . D . n \; [m . s^{-1}] \tag{11}$$

4.2.8 Enthalpy-entropy diagram

Enthalpy entropy diagram or i-s diagram or mollier diagram is derived chard used to plot enthalpy (i) versus entropy (s). this chart contains a series of constant pressure lines, series of constant temperature lines, series of constant dryness factor lines and series of constant superheat lines. This diagram is used for superheated steam and quality is greater than 50%.

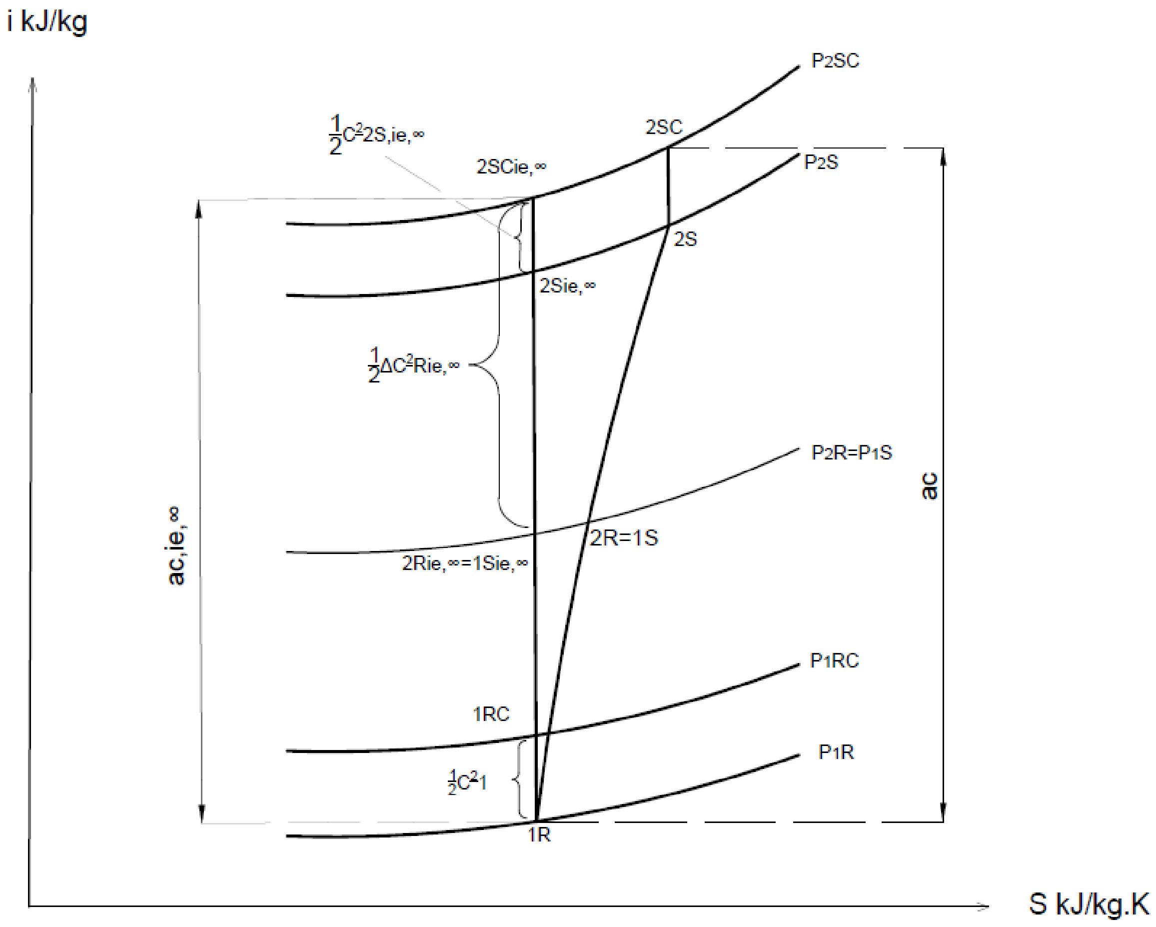


Figure 25 enthalpy-entropy diagram

This curve Figure 25 shows both the curve for the ideal turbo compressor and also for the real turbo compressor.

For an ideal turbo compressor, the values are calculated theoretically and are accurate measures, so they always show us the unique charts but practically there are some losses. Let's describe the chart,

Where

- 1R - This is the place the gas enters the compressor unit. The gas enters the rotor part of the compressor and it contains enthalpy and some kinetic energy. This part is considered to be the inlet or nozzle for the flow of air to the system.
- 1RC - Is the place where the total gas that enters the compressor, which lies on the imaginary isobaric curve p_{1rc}. Where these curves are the constant pressures at

given points. At this point, the air from the inlet or nozzle enters the rotating part of the compressor.

- $2R_{ie,\infty}$ - It is the work at the end of the rotor, the work at the end of the rotor is equal to the work at the start of the stator. From 1R to $2R_{ie,\infty}$ is the impeller where the kinetic energy and the potential energy increases.
- $1S_{ie,\infty}$ -Is the work at the start of the stator. It is the point where the work is transferred from rotor to stator, can be said as the end of the rotor or the start of the stator.
- $2S_{ie,\infty}$ - It is the work at the end of the stator. The point from $1S_{ie,\infty}$ to $2S_{ie,\infty}$ is the diffuser part, where there is a decrease in kinetic energy (velocity decrease) and an increase in the potential energy (pressure energy). According to the ideal process, this point has zero kinetic energy.
- $1/2\Delta c^{2}{}_{, ie,\infty}$ - The difference between the end of the rotor and the end of the stator is the loss of output velocity.
- $1/2\Delta c_2{}^{2}{}_{,s}$ - It is the energy loss at the output by the kinetic energy.

Described below is the process of compression in the ideal centrifugal compressor when it is for the real compression process the infinite symbol ∞ is not used [19].

4.2.9 Real compressor

- The transformation of energy occurs with the direction of an increase in entropy.
- The gas flow through the compressor has experienced internal friction and is not perfectly compressible.
- In the real process, the compression work is an adiabatic process and approaches the polytropic process.
- It has an increase in internal energy and the machines are not tight, have more stages.
- The things continue on both ideal and the real curve is the "equation of total energy" and "velocity triangles".

4.2.10 Advantages of centrifugal compressor

- More reliable and low maintenance.
- Long service life.
- They are oil-free so, compressed gas is not polluted with oil.
- Suitable for continuous compressed air supply.

- Relatively energy efficient.
- Pressure energy increased per stage is higher than pressure increased in axial compressor per stage.
- They have very fewer rubbing parts, achieves low wear of active parts.

4.2.11 Disadvantages of centrifugal compressor

- Higher noise level.
- They are not suitable for very high compression.
- Compared to axial compressors they have a large frontal area.
- They work at high speed; it requires a complex mounting of the vibration.

5 DESIGN AND CALCULATION OF MAIN DIMENSIONS OF RADIAL AIR FAN

For the design and calculation of radial air fan with the given parameters with which the calculations are done. The radial air fan should be designed with a flow performance of 30 [$m^3.s^{-1}$] and a pressure depression 2000 Pa. This is a single-stage fan with one-sided suction with RPM of 500 min^{-1}, and the air temperature at the suction is 40°C and air absolute pressure 1 bar. The fan should be selected such that ideally 50% of the input work is transformed into kinetic energy and 50% of the input work will be used to increase the pressure energy of the gas. The other necessary physical and dimensional parameters are considered appropriately.

Given data		**Unit conversion**
$\dot{V}_d$	30 [m^3*s^{-1}]	[-]
Δp_{zh}	2000 [Pa]	[-]
n	500 min^{-1}	8.33 [s^{-1}]
T	40°C	313.15 K
p	1 bar	100000 [Pa]

Table 1 Given parameters

Useful table variables for air with ideal gas properties:

T	1 [-]
ψ	1.3 [-]
$\psi_{id,\infty}$	2*τ
β_2	90 [°]
c_v	15 [$m.s^{-1}$]
R	287 [$J.kg^{-1}.K^{-1}$]

Table 2 Useful variables

5.1 Density of the intake air

In an ideal gas the pressure p, volume V, and the temperature T are related simply by ideal gas law. The ideal gas law is represented by,

$$pV = nRT \tag{12}$$

Where

- n = m/M is the molar concentration
- p is the pressure

- V is the volume = density*mass
- R is the universal gas constant
- T is the temperature

Hence it can also be written as,

$$pV = mRT \tag{13}$$

The equation for the density of intake air can be represented by,

$$\rho = \frac{p}{r.T} \ [kg.m^{-3}] \tag{14}$$

$$\rho = \frac{100000}{(287) * (313.5)} = 1.113 \ [kg.m^{-3}]$$

5.2 Static part of total specific energy

The static part of total specific energy can be calculated with pressure depression of the fan and the density of the intake air.

$$a_{zh} = \frac{\Delta p_{zh}}{\rho} \ [j.kg^{-1}] \tag{15}$$

$$a_{zh} = \frac{2000}{1.113} = 1796.95 \ [j.kg^{-1}]$$

5.3 Total specific energy

The total specific energy with static and dynamic part of the fan is calculated with the kinetic energy that must be transferred to the air by the fan and the loss of pressure energy or static part of total specific energy. The equation is expressed by,

$$a_c = \frac{c_v^2}{2} + a_{zh} \ [J.kg^{-1}] \tag{16}$$

$$a_c = \frac{15 * 15}{2} + 1796.95 \ [J.kg^{-1}]$$

$$a_c = 1909.45 \ [J.kg^{-1}]$$

It can be seen that after adding the dynamic part of the total specific energy (kinetic energy), only a lower increase of total specific energy is observed.

5.4 Blade shape recapitulation

The blade required to be designed for our design must possess 50% of the added work to increase the pressure energy and 50% of its work to the increase of kinetic energy. The blades give the energy by the value of the square of the peripheral or circumferential velocity.

The peripheral component of the absolute velocity is given by $c_{2u} = u_2$. The blade velocity triangle for the given condition is shown in Figure 26

- **Blade shape factor** – is the ratio of the peripheral or circumferential component of the absolute velocity to the peripheral velocity.

$$\tau = \frac{c_{2u}}{u_2} \tag{17}$$

- **The reaction factor** – represents what part of the externally added work already transforms in the rotor blades in the pressure energy.

$$K = 1 - \frac{1}{2} * \tau \tag{18}$$

- **The work is done** – finally the work that must be added to the machine.

$$a_c = u_2 * c_{2u} \tag{19}$$

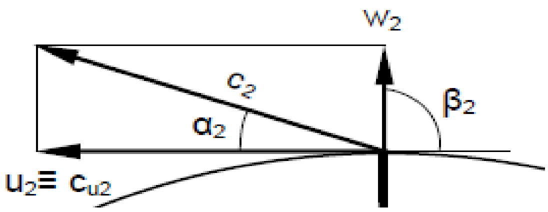


Figure 26 Velocities at radial outlet

The evaluation criteria for the radial outlet blade is given by,

- Blade shape factor, from the equation (17)

$$\tau = \frac{u_2}{u_2} = 1$$

- The reaction factor, from the equation (18)

$$K = 1 - \frac{1}{2} * 1 = \frac{1}{2}$$

- The work from the equation (19)

$$a_c = u_2 * c_{2u} = u_2^2$$

5.5 Peripheral velocity at the outlet from the rotor

The peripheral velocity at the outlet of the rotor is calculated with the help of the pressure factor and the total specific energy.

$$\psi = \frac{a_c}{\frac{u_2^2}{a_c}} \tag{20}$$

Where,

$$u_2 = \sqrt{\frac{2.a_c}{\psi}} \ [m.s^{-1}] \tag{21}$$

$$u_2 = \sqrt{\frac{2 * 1909.45}{1.3}} = 54.2 \ [m.s^{-1}]$$

5.6 Rotor outlet diameter

The rotor outlet diameter of this design can be calculated from the peripheral velocity of the rotor at the outlet from equation (11).

$$u_2 = \pi.D_2.n$$

$$D_2 = \frac{u_2}{\pi.n} \ [m]$$

$$D_2 = \frac{54.2}{\pi.8.33} = 2.071 \ [m]$$

5.7 Volume factor

The volume factor is a dimensionless criterion, which is required to calculate the external diameter of the rotor inlet.

$$\varphi = \frac{\dot{V}_d}{\frac{\pi.D_2^2.u_2}{4}} \ [-] \tag{22}$$

$$\varphi = \frac{30}{\frac{\pi * 2.071^2 * 54.2}{4}} = 0.164 \ [-]$$

5.8 Rotor inlet diameter

The rotor inlet diameter can be calculated by substituting the rotor outlet diameter and the volume factor.

$$\frac{D_1}{D_2} = 1.2 * \sqrt[3]{\varphi} \tag{23}$$

$$D_1 = D_2 * 1.2 * \sqrt[3]{\varphi} \ [m]$$

$$D_1 = 2.071 * 1.2 * \sqrt[3]{0.164} = 1.36\ [m]$$

The inlet diameter of the rotor must be smaller than the outlet diameter of the rotor ($D_1<D_2$) in a centrifugal compressor. So form the calculation the inlet diameter of the rotor is smaller than the outlet diameter of the rotor and the calculation is going correctly.

5.9 Calculation and other parameters required for the construction of velocity triangles

Peripheral velocity at the outlet from the rotor,

$$u_2 = 54.2\ [m.s^{-1}]$$

5.9.1 Absolute velocity at the inlet to the rotor

The absolute velocity at the rotor is calculated from the formula for the volume flow rate.

$$\dot{V}_d = S_1 * c_1 = \frac{\pi * D_1^2}{4} * c_1 \tag{24}$$

The volume flow rate formula is for a single-sided fan. For a fan with a double-sided suction fan, each side intake half the amount of volume flow performance.

$$c_1 = \frac{4 * \dot{V}_d}{\pi * D_1^2}\ \ [m * s^{-1}]$$

$$c_1 = \frac{4 * 30}{\pi * 1.36} = 20.65\ [m * s^{-1}]$$

5.9.2 Relative velocity at the outlet from the rotor

Relative velocity w_2 at the outlet from the rotor is the equal absolute velocity at the inlet to the rotor c_1 because of the radial output of the blade angle $\beta_2 = 90[°]$.

$$w_2 = c_1 \tag{25}$$

$$c_1 = 20.65\ [m * s^{-1}]$$

Which can be also said as when the angle $\beta_2 = 90[°]$, the peripheral component of the absolute velocity is defined as,

$$c_{1u} = c_1 * \cos\beta_2 \tag{26}$$

$$c_{1u} = 20.65 * \cos 90 = 0\ [m * s^{-1}]$$

5.9.3 Peripheral velocity at the inlet to the rotor

The peripheral velocity at the inlet to the rotor is calculated from the equation for the peripheral velocity from equation (11).

$$u_1 = \pi . D_1 . n \ \ [m * s^{-1}]$$

$$u_1 = \pi * 1.36 * 8.33 = 35.59 \ \ [m * s^{-1}]$$

Therefore, the peripheral velocity at the inlet to the rotor u_1 must be smaller than the peripheral velocity at the outlet to the rotor u_2, because the outlet diameter of the rotor is larger than the inlet diameter of the rotor.

5.9.4 Relative velocity at the inlet of the rotor

The relative velocity at the inlet of the rotor can be in two different ways, let's calculate the relative velocity in both the ways.

The calculation by Pythagorean theorem,

$$c = \sqrt{a^2 + b^2} \tag{27}$$

$$w_1 = \sqrt{u_1^2 + c_1^2} \ \ [m * s^{-1}] \tag{28}$$

$$w_1 = \sqrt{35.59^2 + 20.65^2} = 41.15 \ \ [m * s^{-1}]$$

By using “cousins” goniometric function to find the relative velocity at the inlet to the rotor,

$$cos\beta_1 = \frac{u_1}{w_1} \tag{29}$$

Where the angle β_1 is calculated from the “tangent” goniometric function,

$$tan\beta_1 = \frac{c_1}{u_1} \tag{30}$$

$$\beta_1 = 30.12 \ [°]$$

Now the relative velocity at the inlet to the rotor,

$$w_1 = \frac{35.59}{cos30.12} = 41.15 \ [m * s^{-1}]$$

Hence both the method of calculation shows the same result.

5.9.5 Absolute velocity at the outlet from the rotor

The absolute velocity at the outlet of the rotor can be in two different ways, let's calculate the relative velocity in both the ways.

Calculation of absolute velocity at the outlet of the rotor by the Pythagorean theorem, from the equation (27).

$$c = \sqrt{a^2 + b^2}$$

$$c_2 = \sqrt{u_2^2 + w_2^2} \ [m * s^{-1}] \tag{31}$$

$$c_2 = \sqrt{54.2^2 + 20.65^2} = 58 \ [m * s^{-1}]$$

By using "cousins" goniometric function to find the absolute velocity at the outlet to the rotor,

$$cos\alpha_2 = \frac{u_2}{c_2} \tag{32}$$

Where the angle α_2 is calculated from the "tangent" goniometric function,

$$tan\alpha_2 = \frac{w_2}{u_2} \tag{33}$$

$$\alpha_2 = 20.85 \ [°]$$

Now the absolute velocity at the outlet to the rotor,

$$c_2 = \frac{54.2}{\cos 20.85} = 58 \ [m * s^{-1}]$$

5.9.6 Construction of velocity triangles

Summary of velocities and angles of the velocity triangle at the inlet,

- Peripheral velocity at the inlet to the rotor $u_1 = 35.59 \ [m * s^{-1}]$
- Absolute velocity at the inlet to the rotor $c_1 = 20.65 \ [m * s^{-1}]$
- Relative velocity at the inlet to the rotor $w_1 = 41.15 [m * s^{-1}]$
- Angles $\alpha_1 = 90°$, $\beta_1 = 30.12°$

Summary of velocities and angles of the velocity triangle at the outlet

- Peripheral velocity at the outlet from the rotor $u_2 = 54.2 \ [m * s^{-1}]$
- Absolute velocity at the outlet from the rotor $c_2 = 58 \ [m * s^{-1}]$
- Relative velocity at the outlet from the rotor $w_2 = 20.65 [m * s^{-1}]$

- Angles $\alpha_2 = 20.85[°]$, $\beta_2 = 90[°]$

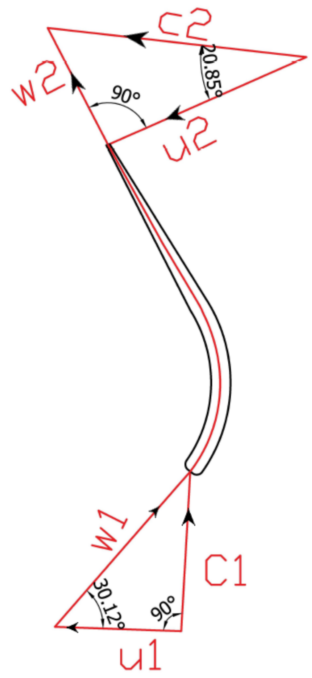


Figure 27 Inlet and outlet velocity triangle form the blade design

5.9.7 Radial fan working and logic of velocity triangle

- The gas enters the inlet to the rotor at absolute velocity c_1.
- To the inlet, the gas passes through the inner diameter D_1 of rotor running at a speed of (RPM) n = 500 [min^{-1}] and the peripheral velocity u_1.
- The inlet to the rotor is always perpendicular to a radial fan at an angle $\alpha_1 = 90[°]$, which is described in the inlet velocity triangle mentioned above, and the longest line in the triangle represents the relative velocity w_1.
- Now the gas in the rotor is in a relatively rotating environment and moves at a relative velocity w, the relative velocity at each point is being tangent to the centerline of the blade profile.
- There is a space between the blades of the rotor, this space between the blades has the shape of diffuser. The diffuser decreases a part of kinetic energy at the cost of pressure energy, this causes a decrease in the relative velocity $w_1 > w_2$.
- After the gas enters the rotor through inner diameter is not the same anymore the dimension of the rotor increases to the outer diameter D_2. Due to this, the peripheral velocity u_2 increases.

- In the rotor, due to acceleration, the absolute velocity at the outlet will be higher than the inlet absolute velocity $c_2 >> c_1$.

5.10 Rotor design

Rotor or impeller is one of the main parts of the design of the radial fan. It helps in the transforming kinetic energy of the gas into pressure energy by reducing the gas velocity.

We have already calculated some of the parameters:

- Rotor inlet diameter D_1 = 1.36 [m]
- Rotor outlet diameter D_2 = 2.071 [m]

So we have the inlet and outlet diameter of the rotor still we need to calculate the width of the rotor. The width of the rotor wheel at the inlet and outlet is calculated from the volume flow rate formula. This formula of volume flow rate is for single-sided suction fans.

$$\dot{V}_d = S_{pl} * c_1 \; [m^3 * s^{-1}] \tag{34}$$

Where the surface of the cylinder shell is calculated by the equation:

$$S_{pl} = \pi * D * b_{ok} \tag{35}$$

Width of the rotor at the inlet,

$$b_{ok,1} = \frac{\dot{V}_d}{\pi * D_1 * c_1} \; [m]$$

$$b_{ok,1} = \frac{30}{\pi * 1.36 * 20.65} = 0.34 \; [m]$$

Width of the rotor at the outlet,

$$b_{ok,2} = \frac{\dot{V}_d}{\pi * D_2 * c_2} \; [m]$$

$$b_{ok,2} = \frac{30}{\pi * 2.071 * 58} = 0.0795 \; [m]$$

5.10.1 Number of blades

The number of blades in the rotor is calculated with the help of an empirical equation.

$$z = \frac{8.5 * sin\beta_2}{1 - \frac{D_1}{D_2}} \; [ks] \tag{36}$$

$$z = \frac{8.5 * sin90}{1 - \frac{1.36}{2.071}} = 25 \ \ [ks]$$

The number of blades calculated may arrive with decimal values. However, it should be rounded to the nearest integer. I propose stainless steel 316 as the production material which has higher corrosion resistance and mechanical properties. The blade is designed with the software SOLIDWORKS 2019 and is drafted using AUTOCAD 2020 is shown in Figure 28 and Figure 29. The width of the rotor is too small for the design, so the width of the blade is adjusted as required for the design.

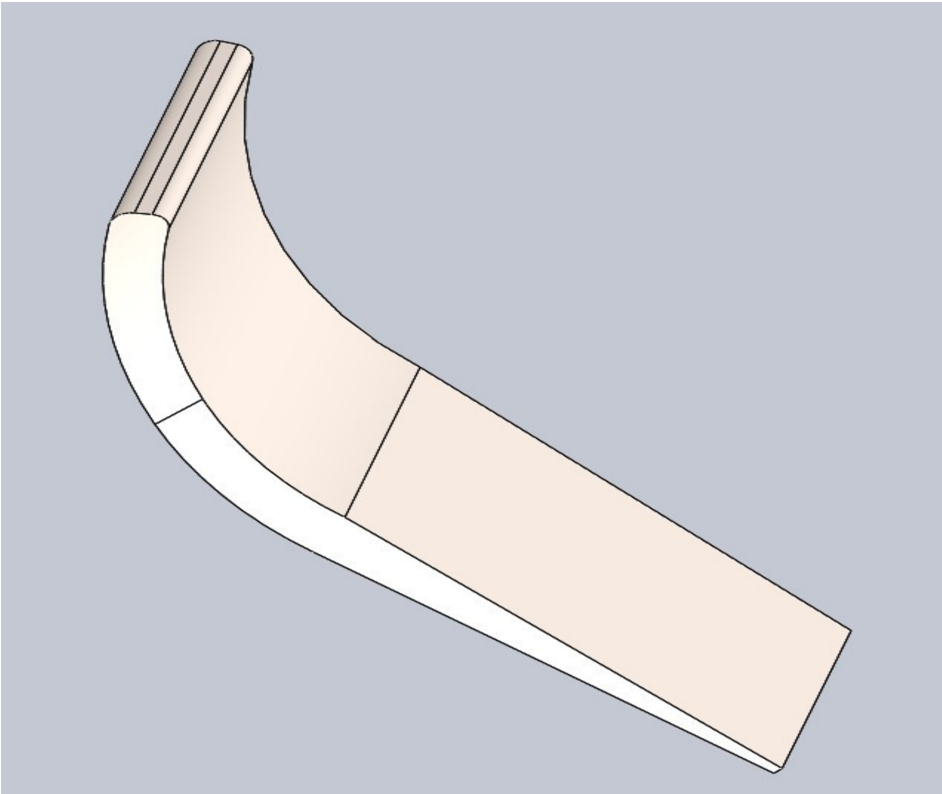


Figure 28 3D design of the blade

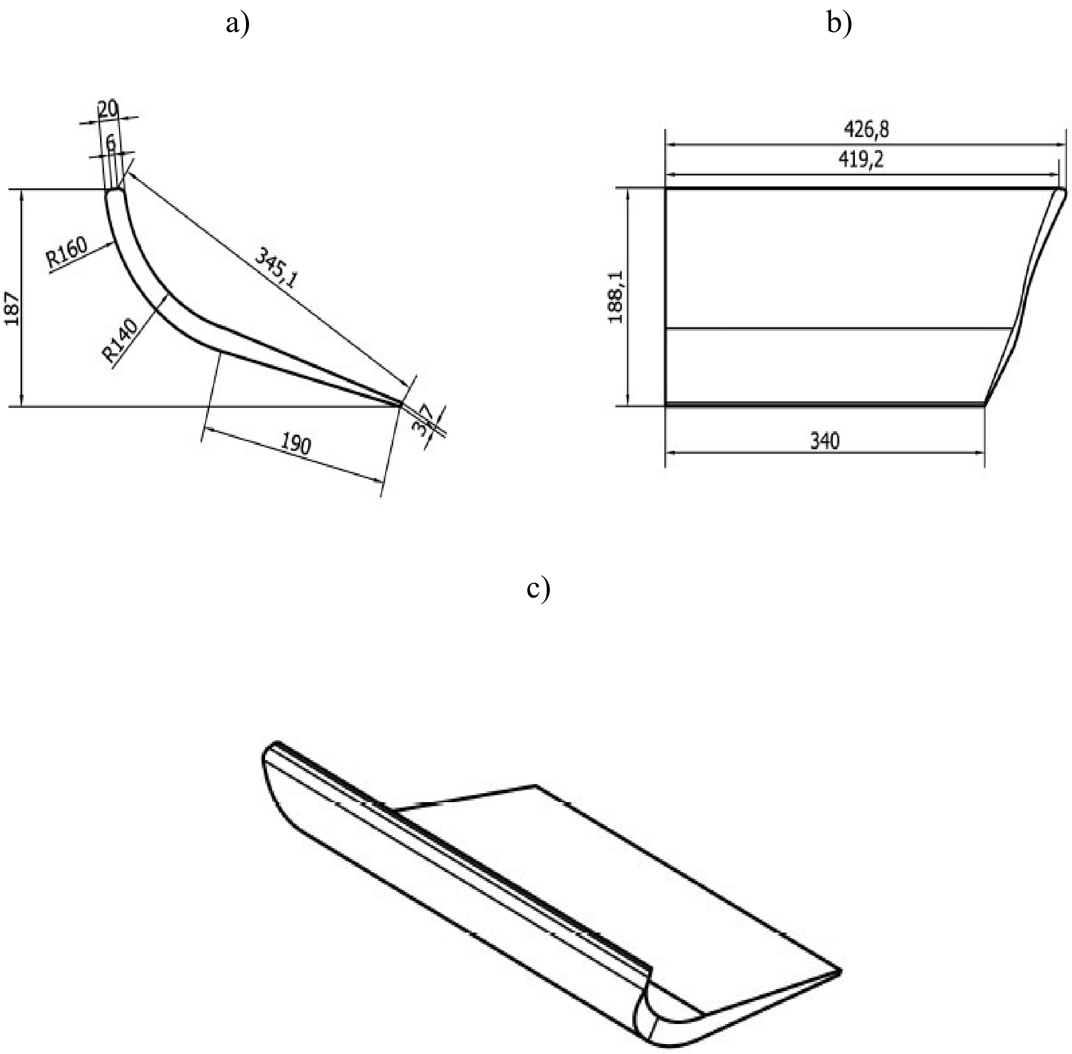


Figure 29 a) Top view, b) Side view, d) Isometric view

5.11 Power consumption of the electric motor

5.11.1 Coefficient constant number of blades

The coefficient constant number of blades can be calculated by the following equation:

$$\varepsilon = 1 - \frac{\pi}{z} * sin\beta_2 \tag{37}$$

$$\varepsilon = 1 - \frac{\pi}{25} * \sin 90 = 0.874\ [-]$$

5.11.2 Hydraulic efficiency

The hydraulic efficiency is calculated by using the ideal pressure factor, pressure factor, and coefficient constant number of blades.

$$\frac{\psi}{\psi_{id,\infty}} = \varepsilon * \eta_h \tag{38}$$

Where it can be rewritten as:

$$\eta_h = \frac{\psi}{\psi_{id,\infty} * \varepsilon} \ [\%]$$

$$\eta_h = \frac{1.3}{2 * 1 * 0.874} = 0.744$$

$$\eta_h = 74.4 \ [\%]$$

5.11.3 Total efficiency

The total efficiency of the electric motor is calculated by the following formula.

$$\eta_c = \eta_h * \lambda * \eta_m \ [\%] \tag{39}$$

The loss factor λ and the mechanical efficiency η_m for the motor is chosen by the choice of the designer.

- The loss factor λ = 0.89 [-]
- mechanical efficiency η_m = 0.95 [-]

$$\eta_c = 0.744 * 0.89 * 0.95 = 0.629$$

$$\eta_c = 62.9 \ [\%]$$

5.11.4 Useful power consumption

The calculation for the useful power consumption is the product of quantity and quality.

$$power\ consumption = quantity * quality \tag{40}$$

$$P_{uz} = \dot{m}_d * a_c \ [W] \tag{41}$$

Where $\dot{m}_d$ the transported amount is calculated by the flow performance $\dot{V}_d$recalculated using the calculated density at the suction side ρ.

$$\dot{m}_d = \dot{V}_d * \rho \ [kg * s^{-1}] \tag{42}$$

$$\dot{m}_d = 30 * 1.113 = 33.39 \ [kg * s^{-1}]$$

Now the useful power consumption is,

$$P_{uz} = 33.39 * 1909.45 = 63756.54[W]$$

5.11.5 Clutch power consumption

The clutch power consumption is calculated equating the total efficiency and the useful power and clutch power consumption.

$$\eta_c = \frac{P_{uz}}{P_{sp}} \ [\%] \tag{43}$$

It can be rewritten as,

$$P_{sp} = \frac{P_{uz}}{\eta_c} \ [W]$$

$$P_{sp} = \frac{63756.54}{0.629} = 101361.75 \ [W]$$

5.11.6 Power consumption of electric motor

The electrical power consumption of the compressor that is the power consumed by the electric motor is calculated from the basic electrical efficiency formula and it is expressed as,

$$\eta_{el} = \frac{P_{sp}}{P_{el}} \ [\%] \tag{44}$$

This equation can be rewritten as,

$$P_{el} = \frac{P_{sp}}{\eta_{el}} \ [W]$$

However, to equate the efficiency the electric motor should be specified and based on the datasheets produced by the manufactures of electric motors it is assumed as,

$$\eta_{el} = 0.945 \ [-]$$

Thus, the power consumption of the electric motor is,

$$P_{el} = \frac{101361.75}{0.945} = 107261.11[W]$$

$$P_{el} = 107.3[kW]$$

5.12 Stator design

5.12.1 Design radius

The design radius is the rotor outlet radius, is calculated by the rotor outlet diameter.

$$r_2 = \frac{D_2}{2} \ [m] \tag{45}$$

$$r_2 = \frac{2.071}{2} = 1.0355 \ [m]$$

5.12.2 Peripheral component of the absolute velocity

The peripheral component of the absolute velocity c_{2u} is identical to the peripheral velocity at the rotor outlet diameter u_2, due to the angle β2 = 90[°].

$$c_{2u} = u_2 \tag{46}$$

$$u_2 = 54.2 \ [m. s^{-1}]$$

5.12.3 Design of spiral chamber

The spiral casing is the housing of the centrifugal fan. The design of the spiral casing is very important, that minimizes the flow loss in the discharge passage. The spiral design is made of 8 different spiral cross-sections.

5.12.4 Equation of logarithmic spirals

To find the radius of the spiral chamber at 8 different parts, the following equation is used.

$$\frac{r_3}{r_2} = e^{\frac{\dot{V}_d^*}{b * c_{2u} * r_2}} \tag{47}$$

Where the design of flow performance $\dot{V}_d^*$ at a given point is calculated by the equation,

$$\dot{V}_d^* = \frac{i}{8} * \dot{V}_d \ [m^3 * s^{-1}] \tag{48}$$

5.12.5 Width of the spiral chamber

In cross-section 0:

The width of the spiral chamber in the cross-section 0 is chosen slightly larger than the width of the rotor.

$$b_0 = b_{ok} \ [m]$$

$$b_0 = 0.35[m]$$

Now the radius $r_{3,0}$ for the design of the spiral chamber an additional of 10mm is chosen from the outer radius of the rotor for clearance.

$$r_{3,0} = 1035.5 + 10 = 1045.5 \ [mm]$$

In cross-section 1:

Width of the spiral chamber,

$$b_1 = 0.385 \ [m]$$

Design flow performance, at the cross-section 1 is calculated from the equation (48).

$$\dot{V}_d^* = \frac{i}{8} * \dot{V}_d \ [m^3 * s^{-1}]$$

$$\dot{V}_d^* = \frac{1}{8} * 30 = 3.75 \ [m^3 * s^{-1}]$$

Design radius of the spiral chamber in cross-section 1, from equation (47).

$$r_{3,i} = e^{\frac{\dot{V}_d^*}{b_1 * c_{2u} * r_2}} * r_2 \ [m]$$

$$r_{3,1} = e^{\frac{3.75}{(0.385)*(54.2)*(1.0355)}} * 1.0355 \ [m]$$

$$r_{3,1} = 1.231 \ [m]$$

In cross-section 2:

Width of the spiral chamber,

$$b_2 = 0.605 \ [m]$$

Design flow performance, at the cross-section 2 is obtained from the equation (48).

$$\dot{V}_d^* = \frac{i}{8} * \dot{V}_d \ [m^3 * s^{-1}]$$

$$\dot{V}_d^* = \frac{2}{8} * 30 = 7.5 \ [m^3 * s^{-1}]$$

Design radius of the spiral chamber in cross-section 2, from equation (47).

$$r_{3,i} = e^{\frac{\dot{V}_d^*}{b_2 * c_{2u} * r_2}} * r_2 \ [m]$$

$$r_{3,2} = e^{\frac{7.5}{(0.605)*(54.2)*(1.0355)}} * 1.0355 \ [m]$$

$$r_{3,2} = 1.291 \ [m]$$

In cross-section 3:

Width of the spiral chamber,

$$b_3 = 0.753 \ [m]$$

Design flow performance, at the cross-section 3 is obtained from the equation (48).

$$\dot{V}_d^* = \frac{i}{8} * \dot{V}_d \ [m^3 * s^{-1}]$$

$$\dot{V}_d^* = \frac{3}{8} * 30 = 11.25 \ [m^3 * s^{-1}]$$

Design radius of the spiral chamber in cross-section 3, from equation (47).

$$r_{3,i} = e^{\frac{\dot{V}_d^*}{b_3 * c_{2u} * r_2}} * r_2 \ [m]$$

$$r_{3,3} = e^{\frac{11.25}{(0.753)*(54.2)*(1.0355)}} * 1.0355 \ [m]$$

$$r_{3,3} = 1.351 \ [m]$$

In cross-section 4:

Width of the spiral chamber,

$$b_4 = 0.875 \ [m]$$

Design flow performance, at the cross-section 4 is obtained from the equation (48).

$$\dot{V}_d^* = \frac{i}{8} * \dot{V}_d \ [m^3 * s^{-1}]$$

$$\dot{V}_d^* = \frac{4}{8} * 30 = 15 \ [m^3 * s^{-1}]$$

Design radius of the spiral chamber in cross-section 4, from equation (47).

$$r_{3,i} = e^{\frac{\dot{V}_d^*}{b_4 * c_{2u} * r_2}} * r_2 \ [m]$$

$$r_{3,4} = e^{\frac{15}{(0.875)*(54.2)*(1.0355)}} * 1.0355 \ [m]$$

$$r_{3,4} = 1.405\ [m]$$

In cross-section 5:

Width of the spiral chamber,

$$b_5 = 0.95\ [m]$$

Design flow performance, at the cross-section 5 is obtained from the equation (48).

$$\dot{V}_d^* = \frac{i}{8} * \dot{V}_d\ \ [m^3 * s^{-1}]$$

$$\dot{V}_d^* = \frac{5}{8} * 30 = 18.75\ [m^3 * s^{-1}]$$

Design radius of the spiral chamber in cross-section 5, from equation (47).

$$r_{3,i} = e^{\frac{\dot{V}_d^*}{b_5 * c_{2u} * r_2}} * r_2\ \ [m]$$

$$r_{3,5} = e^{\frac{18.75}{(0.95)*(54.2)*(1.0355)}} * 1.0355\ \ [m]$$

$$r_{3,5} = 1.471\ [m]$$

In cross-section 6:

Width of the spiral chamber,

$$b_6 = 1.04\ [m]$$

Design flow performance, at the cross-section 6 is obtained from the equation (48).

$$\dot{V}_d^* = \frac{i}{8} * \dot{V}_d\ \ [m^3 * s^{-1}]$$

$$\dot{V}_d^* = \frac{6}{8} * 30 = 22.5\ [m^3 * s^{-1}]$$

Design radius of the spiral chamber in cross-section 6, from equation (47).

$$r_{3,i} = e^{\frac{\dot{V}_d^*}{b_6 * c_{2u} * r_2}} * r_2\ \ [m]$$

$$r_{3,6} = e^{\frac{22.5}{(1.04)*(54.2)*(1.0355)}} * 1.0355\ \ [m]$$

$$r_{3,6} = 1.52\ [m]$$

In cross-section 7:

Width of the spiral chamber,

$$b_7 = 1.088 \; [m]$$

Design flow performance, at the cross-section 7 is obtained from the equation (48).

$$\dot{V}_d^* = \frac{i}{8} * \dot{V}_d \;\; [m^3 * s^{-1}]$$

$$\dot{V}_d^* = \frac{7}{8} * 30 = 26.25 \; [m^3 * s^{-1}]$$

Design radius of the spiral chamber in cross-section 7, from equation (47).

$$r_{3,i} = e^{\frac{\dot{V}_d^*}{b_7 * c_{2u} * r_2}} * r_2 \;\; [m]$$

$$r_{3,7} = e^{\frac{26.25}{(1.088)*(54.2)*(1.0355)}} * 1.0355 \;\; [m]$$

$$r_{3,7} = 1.591 \; [m]$$

In cross-section 8:

Width of the spiral chamber,

$$b_8 = 1.145 \; [m]$$

Design flow performance, at the cross-section 7 is obtained from the equation (48).

$$\dot{V}_d^* = \frac{i}{8} * \dot{V}_d \;\; [m^3 * s^{-1}]$$

$$\dot{V}_d^* = \frac{8}{8} * 30 = 30 \; [m^3 * s^{-1}]$$

Design radius of the spiral chamber in cross-section 8, from equation (47).

$$r_{3,i} = e^{\frac{\dot{V}_d^*}{b_8 * c_{2u} * r_2}} * r_2 \;\; [m]$$

$$r_{3,8} = e^{\frac{30}{(1.145)*(54.2)*(1.0355)}} * 1.0355 \;\; [m]$$

$$r_{3,8} = 1.651 \; [m]$$

Summary of the stator design for all the cross-section is shown in Table 3:

Cross-section “I”	Width of the spiral chamber b_i [m]	Design radius $r_{3,I}$ [m]
0	0.35	1.0455
1	0.385	1.231
2	0.605	1.351
3	0.753	1.351
4	0.875	1.405
5	0.95	1.471
6	1.04	1.52
7	1.088	1591
8	1.145	1.651

Table 3 Stator design cross-section

However, the housing is made as 3 parts as front plate, backplate, and connector. I propose stainless steel 316 as the production material to avoid corrosion. The housing is surface welded, the design is made in SOLIDWORK 2019 and 2D scheme is made in AUTOCAD 2020 is shown in Figure 30 and Figure 31.

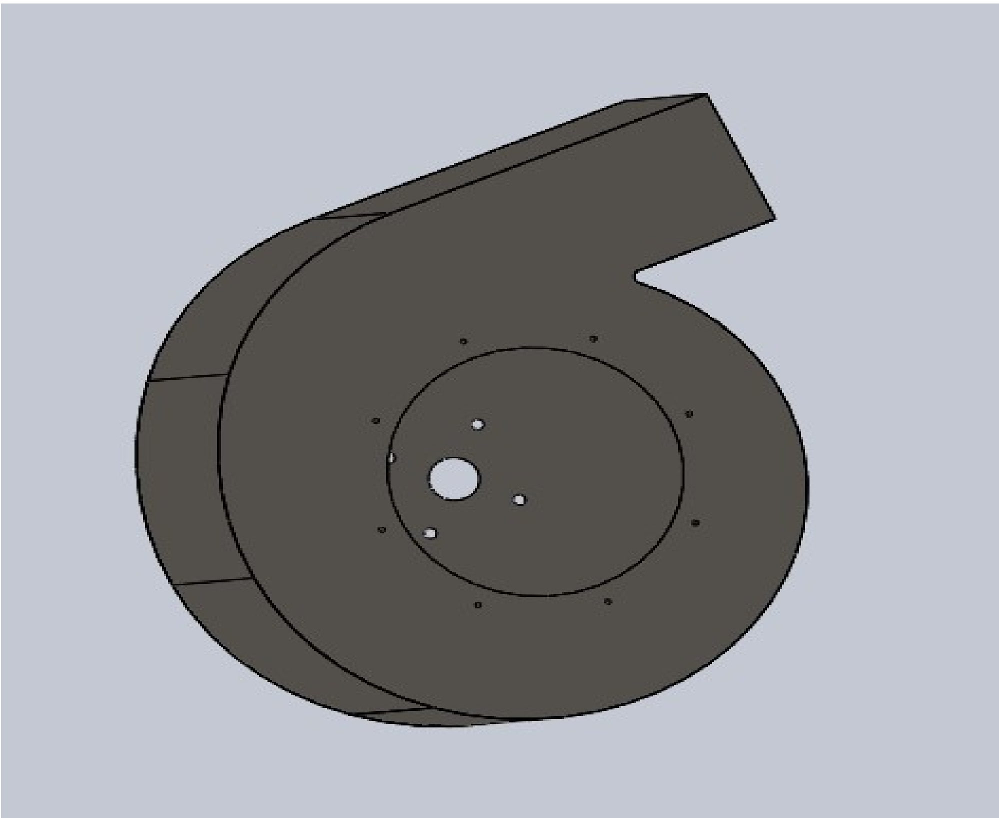


Figure 30 3D design of housing

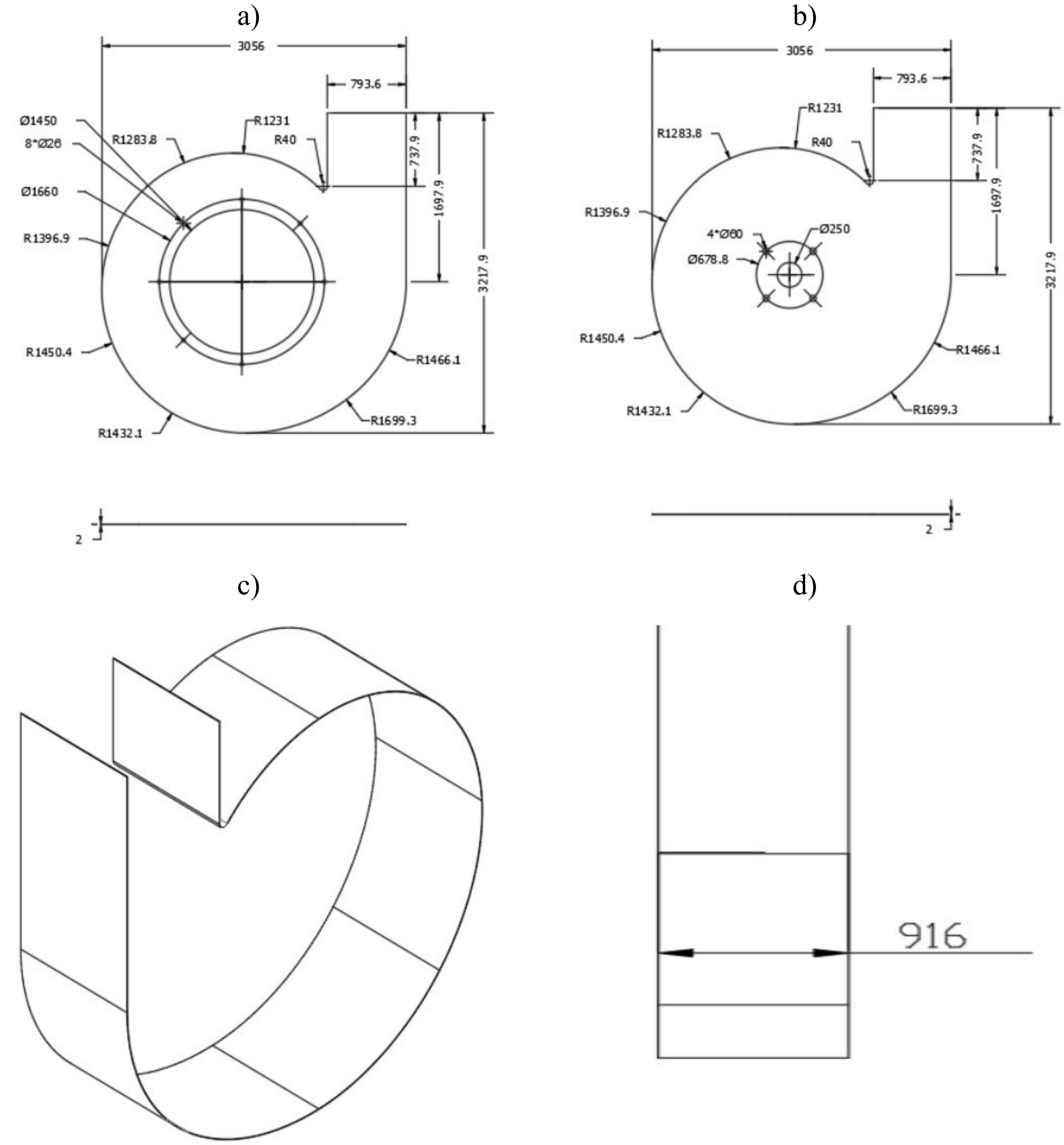


Figure 31 a) Housing front plate, b) Housing backplate, c) Housing connector isometric view, & d)Connector top view

5.13 Shaft

The designed impeller wheel is mounted on the shat. The shat is the mechanical component connected with the impeller and the motor and transmits the drive torque delivered from the motor. The shaft diameter is selected according to ANSI Standard B17.1 for keys and key seats. The shaft is designed with 240 [mm] diameter, the key 56×32 [mm] (b×d) and the keyway with width 56 [mm] and depth 12.4 [mm] [17].

5.14 Bearing

Bearing is a mechanical part that helps the shaft to rotate smoothly. I chose thrust ball bearing according to DIN 711:1988 and DIN 715:1987 and selected 51148 M bearing of size 240x300x45mm. These bearings are used in industrial fans, wind electricity, mining, etc.

A suitable square flange is also designed for the bearing and is fixed to the housing of the radial fan through fasteners.

5.15 Fasteners

The inlet cone and the square flange bearing are fixed to the housing with the help of fasteners. I have chosen different sizes for inlet and square flange. Fasteners are selected according to DIN 931 (ISO 4014) standards hexagon head bolts of M24 with 24 x 75 [mm] dimension for inlet cone and M60 for fixing square flange to the housing.

5.16 Assembly

The calculation is done with finding the rotor and stator design parameters, the remaining parameters are chosen according to the requirement of the designs. Stainless steel 316 is proposed for rotor and stator design, as it could withstand high corrosion. All the parts were committed to surface welding. The electric motor is presented by a schematic view. The 3D design of the fully designed radial fan is shown in Figure 32 and Figure 33.

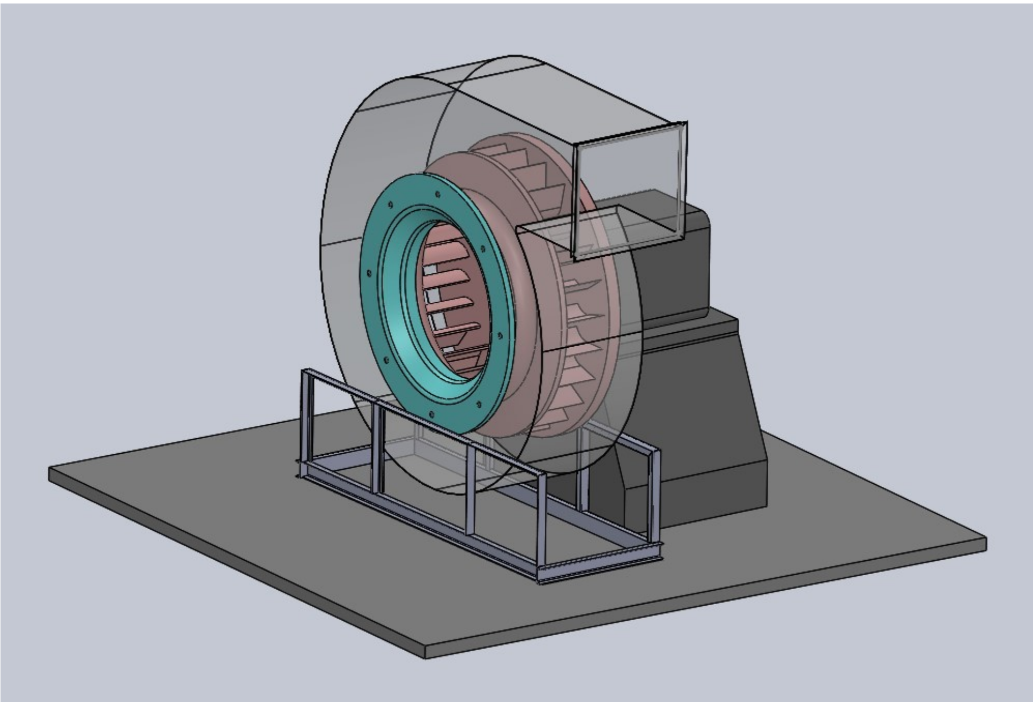
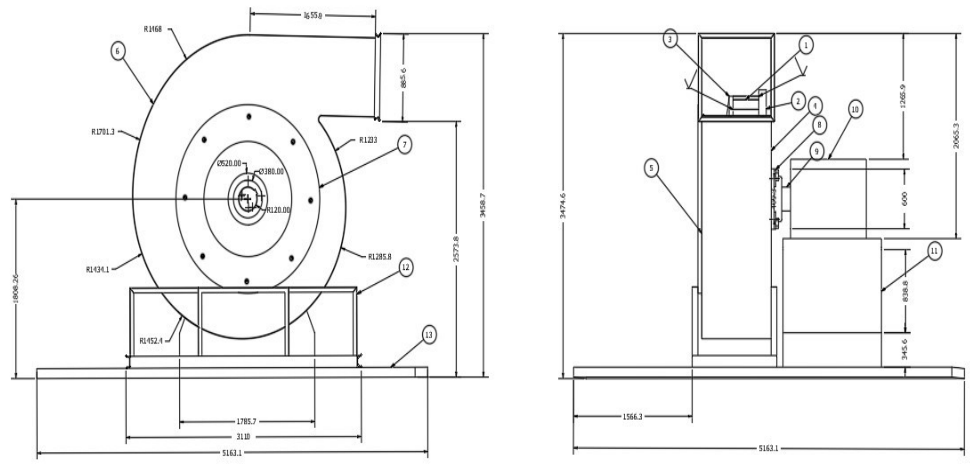
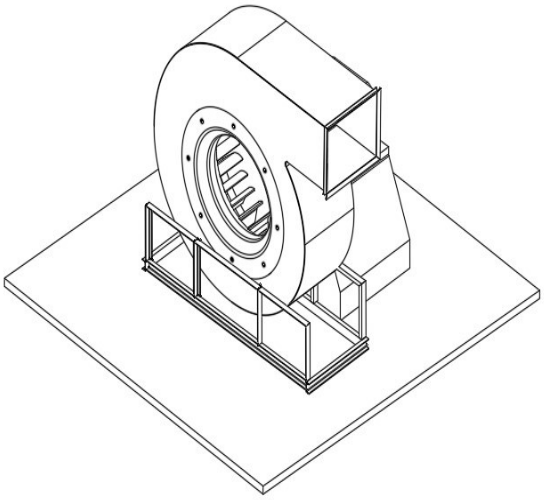


Figure 32 3D Assembly design





S.no	QUANTITY	PART NAME	MATERIAL	WEIGHT	TOLERANCE	EN form./norms
1	25	Blade	1.4401	17.2	ISO 2968 mH	
2	1	Rotor backplate	1.4401	2080.5	ISO 2968 mH	-
3	1	Rim	1.4401	1003.2	ISO 2968 mH	-
4	1	Housing backplate	1.4401	91.3	ISO 2968 mH	
5	1	Housing frontplate	1.4401	92.5	ISO 2968 mH	-
6	1	Housing plate connector	1.4401	103.75	ISO 2968 mH	
7	1	Inlet cone	Alloy 61s	306.6	ISO 2968 mH	-
8	1	Square flange	1.4401	171	ISO 2968 mH	-
9	1	Shaft	1.4401	291	ISO 2968 mH	
10	1	Motor	-	-	ISO 2968 mH	-
11	1	Motor base	Concrete	-	ISO 2968 mH	-
12	1	Stand	1.4401	175	ISO 2968 mH	
13	1	Base	Concrete	-	ISO 2968 mH	-
14	8	Nut	1.4401		ISO 2968 mH	BS EN ISO 4017 Metric M24X75mm
15	8	Bolt	1.4401	-	ISO 2968 mH	ISO 4033-M24
16	8	Washer	1.4401	-	ISO 2968 mH	DIN EN ISO 7092 -M24
17	4	Nut	1.4401		ISO 2968 mH	BS EN ISO 4017 Metric M60X75mm
18	4	Bolt	1.4401	-	ISO 2968 mH	ISO 4033-M60
19	4	Washer	1.4401	-	ISO 2968 mH	DIN EN ISO 7092 - M60

Name	KANNAN K.G	VŠB-TUO	Thesis title: DESIGN OF A INDUSTRIAL RADIAL AIR FAN
Date	01/01/2020		
Tolerance	ISO 2968-mH	Sheet A4	Part title: ASSEMBLY
Weight	5055.72		
Material	1.4401	Scale 1:13	Poz: N/A
Quantity	1		

Figure 33 Drafting of Industrial Radial Air Fan

5.16 Selection of electric motor

From the result of electric power consumption obtained during the calculation is 107.3 [kW], since it is important to propose a suitable electric motor that fulfills the technical parameters of the centrifugal fan. From Siemens, an electric motor is suggested for this designed industrial radial fan. The electric motor suggested here is the SIMOTICS SD 1LE1, which is made of cast iron housing. These motors are designed in a way to handle an extremely rugged, harsh environmental condition, also prevent from dust, vibration, and corrosive atmosphere in the petrochemical industry [22].

Figure 34 Electric motor

This motor lies in the international efficiency standard in the class IE2, which gives high efficiency. 1LE1 are popular for their extremely long life the weight-optimized design that offers stability in the equipment units, more flexible, and are user friendly. This suggested motor is used here because the energy efficiency class is in IE2, thus the motor comes under the efficiency classification according to categories of IEC 60034-30-1:2014.

The motors are available in two different specifications like basic line and performance line to serve with a wide range of specifications. Both specifications are explained clearly in the following Table 4.

Function	**Basic line**	**Performance line**
Bearing size	62 (63 from shaft height 280)	63
Relubrication	Optional	Standard
Paint system	Standard coating, corrosion class C2	Special coating, corrosion class C3
Rating plate	Plastic	Steel
Fan cover	Plastic	Steel
warranty	12 months	36 months

Table 4 Basic and performance line motor specification

This motor uses a 1LE1 series terminal box, which is divided diagonally and also can be rotated 4x through 90°. This helps in connecting the motor cable in any direction. This helps in easy installation and reduces working time. Therefore, the technical specification of the suggested motors with cast iron casing is as follows in Table 5:

Mechanical data	
Size	280M
Power range	84.01[kW] to 123 [kW]
Number of poles	2,4,6,8
Efficiency class	IE2 – High efficiency
Direction of rotation	Bidirectional
Frame material	Cast iron
Method of cooling	IC411 – self ventilated, surface cooled
Insulation	155(F) to 130 (B)
Ambient temperature	-20°C to 40°C
Vibration severity grade	A
Frequency	60Hz
Weight	650 kg

Table 5 Technical data of the electric motor

6 CONCLUSIONS

In this thesis, I have described the theory part of machines of energy transformation their types with examples. A short description of the compressor and their types according to the categories are discussed. Then the brief description of the dynamic compressor their types including fans and centrifugal compressor which has more concentration in theory. The explanation of the centrifugal compressor includes in describing the working process, stages of the compressor, working process with enthalpy entropy diagram, main parts of the centrifugal compressor, and their advantages and disadvantages are explained.

The second part of the thesis includes the calculation and design of the industrial radial fan. In the calculation, the main dimensions like a blade, rotor, and stator values are calculated. Power consumption of the electric motor is also calculated, and I have selected a suitable electric motor for the demand for power consumption. I suggested SIMOTICS SD 1LE1 from siemens, which is a severe duty motor considering the industrial usage application. The manufacturing design is made according to the calculation. The values in the design will not suites perfectly in the design, so required modifications are done to obtain the design. The design will not end up with designing the calculated dimensions some parts other than the calculated values are selected according to the requirements of the design. The design software SOLIDWORKS 2019 is used for designing and AUTOCAD 2020 is used for 2D schemes, also the assembly of the radial fan and the suitable material for the parts is suggested. Finally, the design of an industrial radial air fan done successfully.

7 REFERENCES

[1] windmill energy transformation - Noktan.snaptier.co, 2020. *Noktan.snaptier.co* [online],

[2] CONTRIBUTORS, TMI and CONTRIBUTORS, TMI, 2020, AIR AND GAS FLOW MEASUREMENT - Turbomachinery Magazine. *Turbomachinery Magazine* [online]. 2020. [Accessed 10 May 2020]. Available from: https://www.turbomachinerymag.com/air-and-gas-flow-measurement/

[3] Heat exchanger, 2020. *En.wikipedia.org* [online],

[4] BREZGIN, D. V., ARONSON, K. E., MAZZELLI, F. AND MILAZZO, A., The surface roughness effect on the performance of supersonic ejectors. *Thermophysics and Aeromechanics*. 2017. Vol. 24, no. 4, p. 553-561. DOI 10.1134/s0869864317040060. Pleiades Publishing Ltd

[5] Boiler Feed Injector Diagram.svg - Wikimedia Commons, 2020. *Commons.wikimedia.org* [online],

[6] Fans and Blowers Module V., 2020. *Slideplayer.com* [online],

[7] Brushless Blowers / Pumps | Domel, 2020. *Domel.com* [online],

[8] Axial Compressors - an overview | ScienceDirect Topics, 2020. *Sciencedirect.com* [online],

[9] STEWART, M., 2018, *Surface Production Operations: Volume IV: Pumps and Compressors*. 1st Edition, p. 527-653. Gulf Professional Publishing.

[10] 2020. *Iitism.ac.in* [online],

[11] COPCO, A., 2020, Dynamic Compressors: Centrifugal and Axial Compressors Atlas Copco. *Atlas Copco* [online]. 2020. [Accessed 10 May 2020]. Available from: https://www.atlascopco.com/bg-bg/compressors/wiki/compressed-air-articles/dynamic-compressors

[12] BASIC DESIGN OF CENTRIFUGAL FAN, 2020. *Centrifugalfanengineering.blogspot.com* [online],

[13] Kapsco Moto NEW Portable Mini Air Compressor Electric Tire Inflator Pump 12 Volt Car 12V PSI | Walmart Canada, 2020. *Walmart.ca* [online],

[14] ROCK DRILL SALES & SERVICE, INC., 2020, Atlas Copco Mobile Air Compressors | Rock Drill Sales. *Rockdrillsales.com* [online]. 2020. [Accessed 10 May 2020]. Available from: https://www.rockdrillsales.com/onsite-equipment/portable-air-compressor/

[15] Fan Types – Why Choose a Backward Curved Centrifugal fan?, 2020. *Rs-online.com* [online],

[16] The Compressed Air Blog, 2020. *The Compressed Air Blog* [online],

[17] 2020. *Micro-machine-shop.com* [online],

[18] 2020. *Systemair.com* [online],

[19] YAHYA, S. M., 2013, *Turbines compressors and fans*. 2. p. 517-557. New Delhi: Tata McGraw-Hill Education.

[20] Fan Types - Why choose a forward curved centrifugal fan, 2020. *Rs-online.com* [online],

[21] FAN / EXHAUST / RUGGED / INDUSTRIAL, 2020, IEPW / IEFB - Radial fan by AirPro Fan & Blower | DirectIndustry. *Directindustry.com* [online]. 2020. [Accessed 11 May 2020]. Available from: https://www.directindustry.com/prod/airpro-fan-blower/product-157440-1639497.html

[22] 2020. *Mall.industry.siemens.com* [online],

[23] ŠMÍDA, Zdeněk. *Dynamic compressors with emphasis to the bladed machines,* Lecture No. 2 in the Compressors, Fans and Pumps subject. VSB-Technical University of Ostrava, 2019. 81 p. Internal teaching material

[24] ŠMÍDA, Zdeněk. *Dynamic compressors with emphasis to the bladed machines,* Lecture No. 6 in the Compressors, Fans and Pumps subject. VSB-Technical University of Ostrava, 2019. 98 p. Internal teaching material

[25] ŠMÍDA, Zdeněk. *Dynamic compressors with emphasis to the bladed machines,* Practice No. 10 in the Compressors, Fans and Pumps subject. VSB-Technical University of Ostrava, 2019. 31 p. Internal teaching material

LIST OF ANNEXURES